AD 623840

Final Report
F-B1983

Technical Report

MANUAL BLOWER DEVELOPMENT

by

OFFICE OF CIVIL DEFENSE
OFFICE OF RESEARCH
THE PENTAGON, WASHINGTON, D.C.

Under

Contract No. OCD-OS-62-280

by

Géza Vermes
George Peter Wachtell

September 1965

*This Report has been reviewed in the Office of Civil Defense
and approved for publication. Approval does not signify
that the contents necessarily reflect the views and
policies of the Office of Civil Defense.*

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Work Unit 1432A

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ABSTRACT

This report presents the method of selecting a suitable blower type for manual drive; describes the experimental and theoretical investigations which led to the development of three key components (valves, piston and surge-tank). Results of tests are analyzed that show the blower having more than 65% total efficiency and a stiff pressure vs volume curve. Design for mass production is suggested; calculation of components for the mass produced version is explained so that the mass produced blower should at least be as efficient as the test-version described in this report.

SUMMARY

Whereas the capacity of centrifugal and axial flow blowers is quite sensitive to changes in pressure differential, the capacity of positive-displacement blowers of either the reciprocating or rotary types is quite constant at a given speed. If clearance volumes and seal leakage rates are minimized in the design, positive-displacement blowers serve also as flow meters, with free air delivery essentially dependent upon speed alone.

Blowers having such performance characteristics would be particularly applicable in the ventilating systems of special shelters in which

    a. The net area of air intake and exhaust openings is restricted, or protective apparatus having a relatively high resistance to air flow is included in the ventilating system,

    b. there is a requirement for maintenance of a pressure slightly above atmospheric in the shelter,

    c. the need for fresh air is minimal because environmental cooling is accomplished by means other than ventilation with outside air; that is, by earth conduction effects, well water, a stored cooling agent or mechanical refrigeration, and/or

    d. the electric power supply may be disrupted.

Although the relatively high pressure and low volume flow puts this machine on the specific-speed vs specific-diameter chart into a region

$$(N_S = 1.7, \ D_S = 3.55)$$

where 20% - 30% efficiencies are common (see Fig. 3 of Appendix 1), development work on the components of this manual blower resulted in a design that has more than 65% overall efficiency. It has a pressure-volume curve that displays less than 25% reduction in delivery rate for the entire pressure range. The design has no unstable mode of operation. It has no preferred rotational direction. The machine has no high-speed precision-manufactured components (such as expensive gears).

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1.0 INTRODUCTION

The scope of the work to be performed under Contract No. OCD-OS-62-280 (June 30, 1962) can be outlined as follows:

1) A comparative evaluation of the characteristics of various types of blowers and drive mechanisms;

2) A study of human factors involved in operating blowers with relation to shelter ventilating requirements;

3) The preparation of criteria including designs and specifications for at least one selected size and type of combination blower unit;

4) The fabrication and delivery to the Government of a prototype unit of the preferred type;

5) The determination of prototype performance by means of appropriate laboratory tests;

6) The final modification of design specifications and prototype unit.

Specifications were as follows:

The prototype blower unit should have a rated capacity within the range of 150-300 cfm to be used either singly or in multiple in community shelters; it should be capable of developing six inches of water static pressure at rated capacity with motor drive, and, within the limits of human energy expenditure, with manual drive. During the course of the project it was further stipulated that the blower should develop 4.5 in. of w. static pressure with manual drive, and it should have a steep pressure vs volume characteristic curve (i.e. large increase in pressure should correspond to small reduction in flow).

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Considering the objectives stated above it is obvious that the study of human factors involved in operating blowers in shelters could not be more than preliminary without the blower being available. It was decided, therefore, that selection of the type of blower should be done by using available information on human factors; the quantitative appraisal of the influence of shelter environment should be done <u>after</u> the blower will be available.

The available information on human factors suggested two additional stipulations for the blower:

1) the maximum power that could be expected from a person for a reasonable length of time;

2) the most convenient way of delivering this performance.

It is shown in Appendix 1 that 0.1 HP is a comfortable level for pedaling, but also 0.12 - 0.13 HP can be obtained without much stress from the average person in a bicycling position. Considering the above mentioned 150 cfm at 4.5 in.w. requirement, 80% blower efficiency would be necessary with 0.13 HP performance. This performance, however, cannot be expected at very high speeds. 55 rpm is a reasonable speed for the human operator.

2.0 SELECTION OF THE TYPE OF THE BLOWER

The above requirements being fixed, essentially two ways are open to select the type of blower:

1) either consider preferably <u>all</u> designs on the market, or

2) try to narrow down the choice on the basis of theoretical considerations, by using appropriate parameters that will enable the quantitative comparison of different types of blowers; the final choice will thus be narrowed down to a very few types. The selection was done and reported

in Appendix 1; an explanation and justification of the method used (specific speed and specific diameter as artificial parameters), and a brief description of the mathematical background is also contained in Appendix 1.

The type selected was the positive displacement blower. Design speed: 150 rpm at 150 cfm; this design enables the use of well developed standard components: bicycle drives, transmissions, speed changers, etc. The delivery rate of positive displacement machines is relatively insensitive to changes in pressure and their characteristic curve does not have an unstable region - as opposed to most of the hydrodynamic type machines that would have been considered. As the application of several blowers in parallel should be made possible the stable characteristic curve must be considered as a significant feature.

In order to keep the losses as low as possible, instead of a rigid piston, with its inherent friction losses, a pliable diaphragm was first considered. The necessary diameter for this design was 2 feet, a rather bulky machine (Fig. 1). Two other conceptual designs were worked out as alternatives to have some comparison before the final choice has been made: a single stage (Fig. 2) and a two-stage centrifugal blower (Fig. 3). The foil-type blades on the single stage, 3000 rpm (N_s = 104) machine are meant to show that if this machine would be built, all possible aerodynamic refinement has to be applied, as the required pressure cannot be met at the design point of the blower. It has to be overdesigned and operated at a lower than design volume (Fig. 4). This off-design operation reduces the efficiency from the design-value. It can also be seen from Fig. 4 that the pressure margin is extremely low. The two-stage, concentric, double impeller design shown on Fig. 3 attempts to obtain the required pressure without resorting to a very low specific speed. The head is divided between the two runners in the ratio of 2:1, the inner, 3300 rpm impeller delivering the larger portion. The counterrotating 3000 rpm outer runner acts as the collector for the inner wheel. Considering the

- 3 -

7" STROKE

YOKE

LINEAR BALL BEARING

FLY WHEEL

SPEED CHANGER

ROLLER

2' DIAM.

LINEAR BALL BEARING

INLET VALVE

4"

1' DIAM.

PISTON ROD

DIAPHRAGM

RETAINER

INLET VALVE

FIG. I DIAPHRAGM BLOWER

MOTOR

BASE PLATE

CUT A-A

12" DIAM.

CHAIN DRIVE
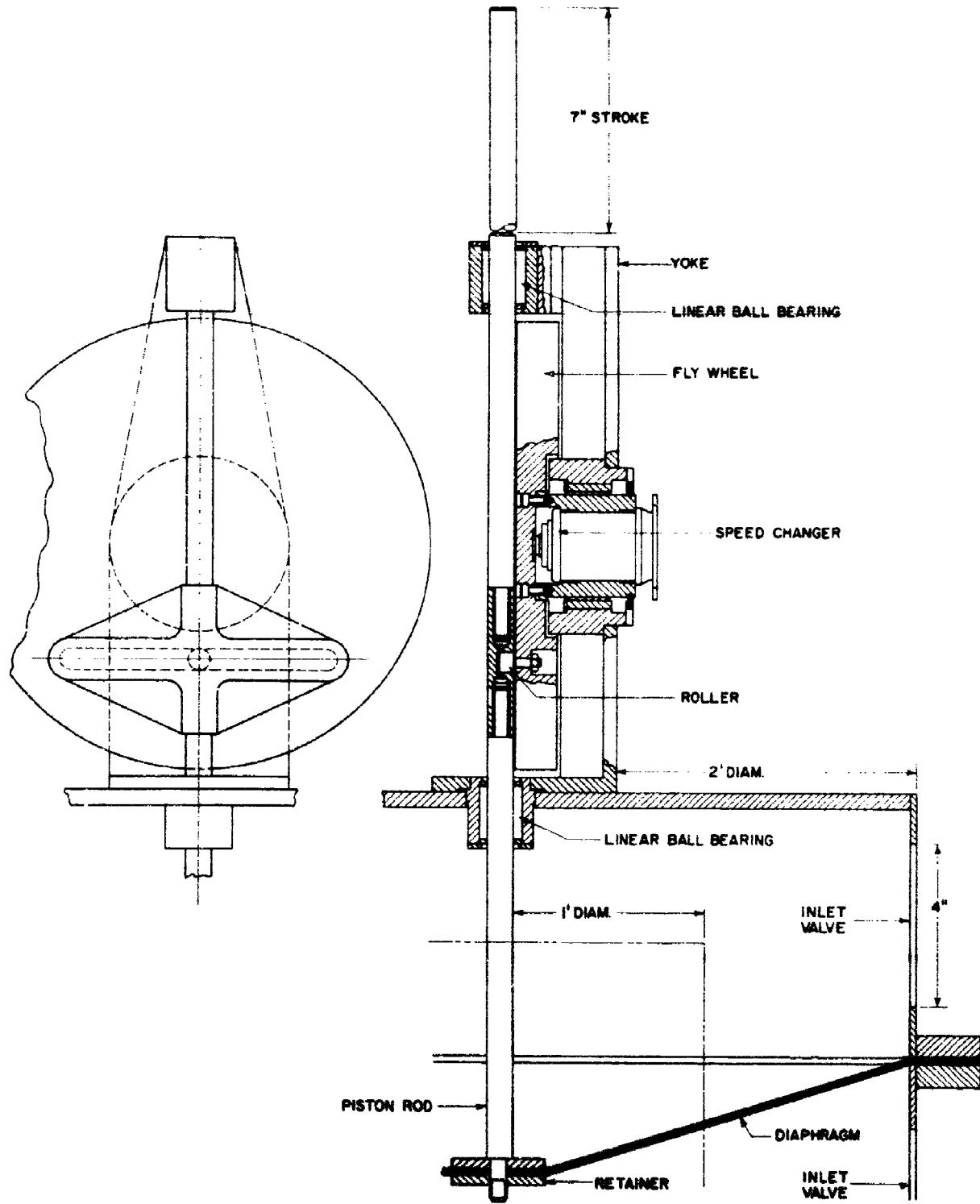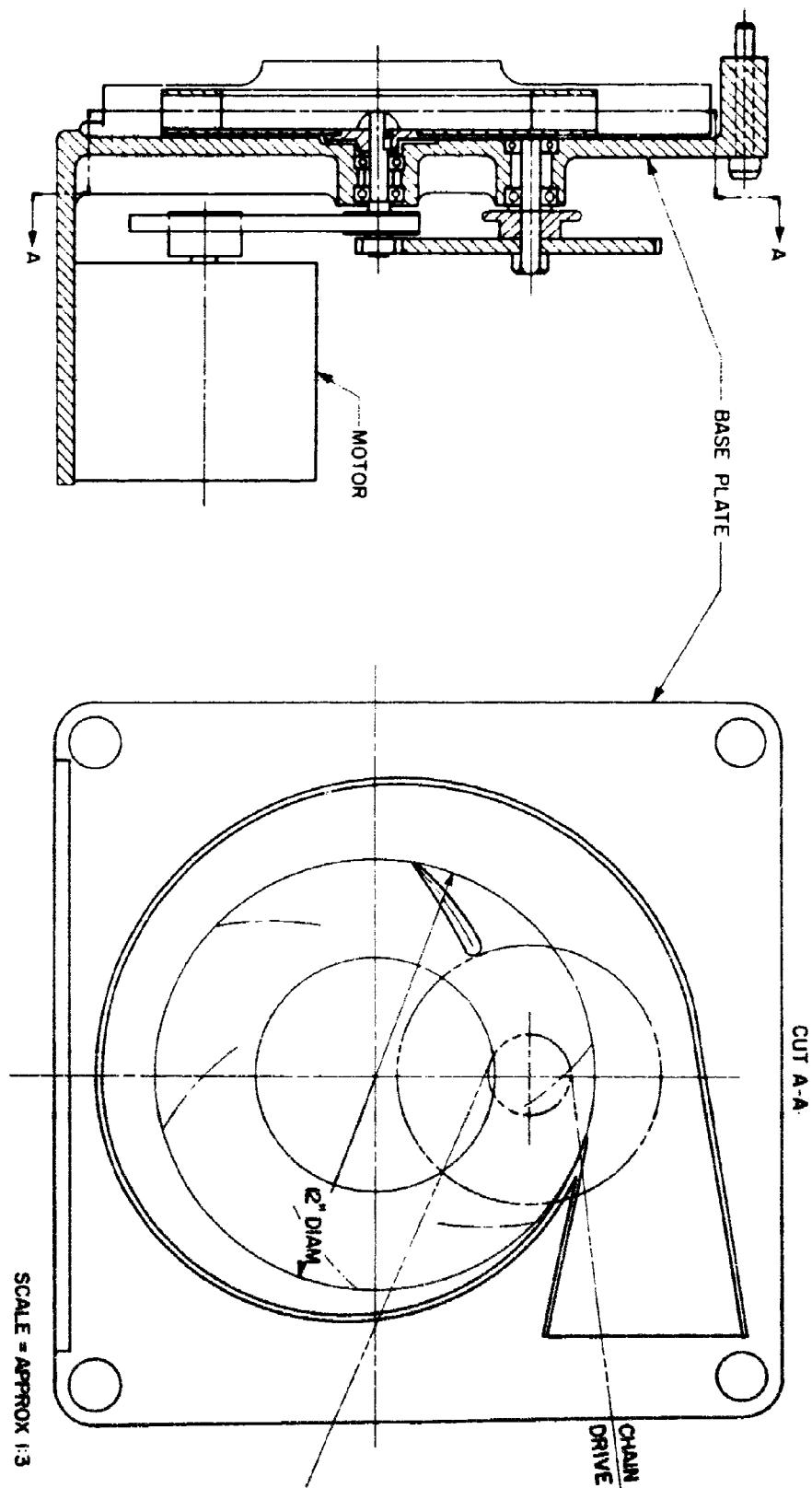
FIG. 2 SINGLE STAGE CENTRIFUGAL BLOWER

SCALE = APPROX 1:3

INNER IMPELLER

OUTER IMPELLER

MOTOR

9"

15 3/4"

4.00"

2.65"

24"

SEAT ₵

20" TO TOP OF SEAT

PEDAL

PEDAL
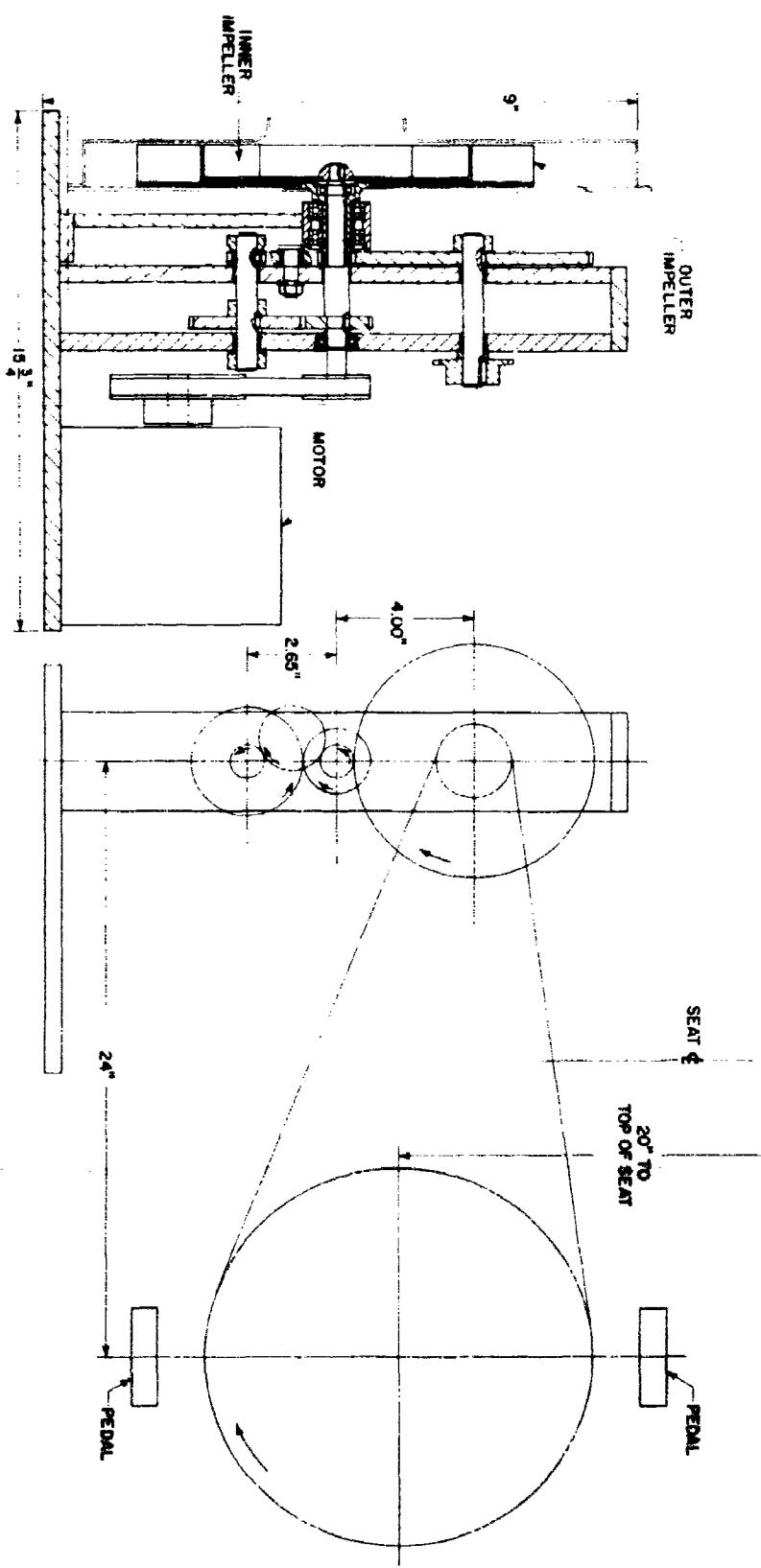
FIG. 3 TWO STAGE CENTRIFUGAL BLOWER
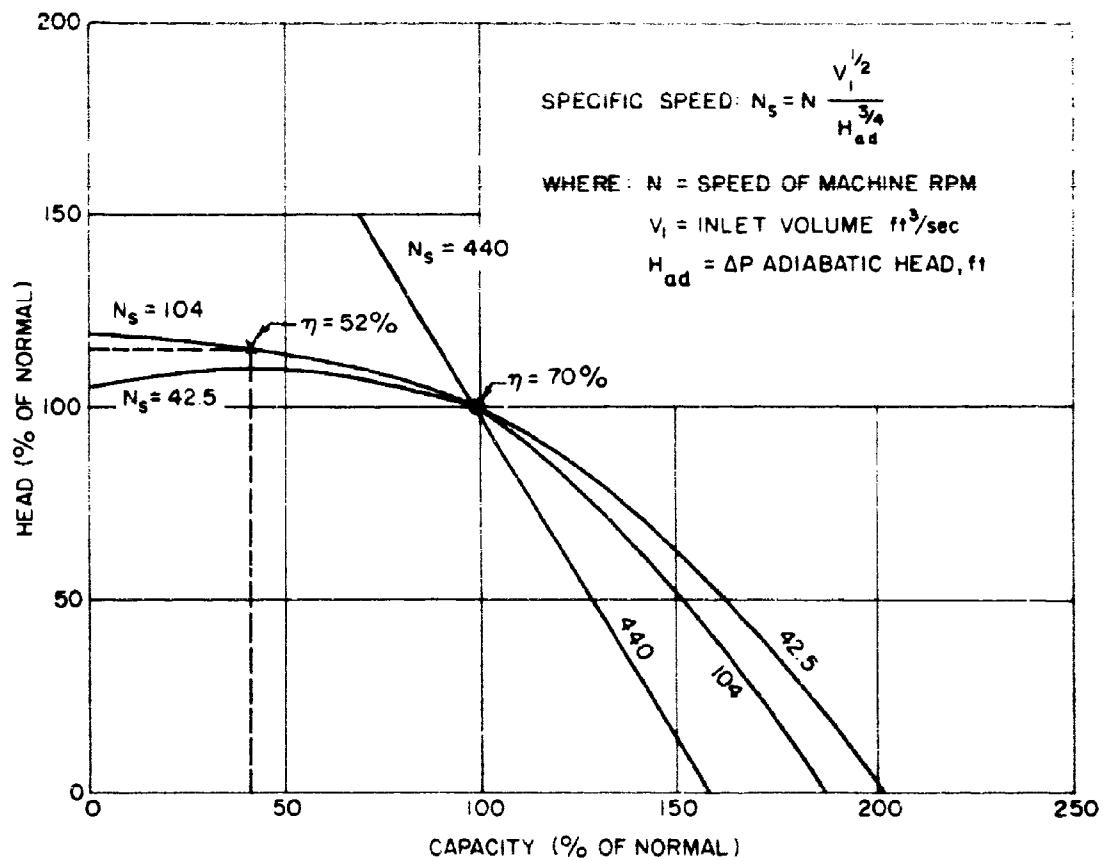
FIG. 4. HEAD-CAPACITY CURVES FOR SEVERAL SPECIFIC SPEEDS

(Source: A.F. Stepanoff, Centrifugal and Axial Pumps. Fig 9-1)

complicated design of the counterrotating wheels, the more favorable specific speed (N_s = 90) and thus somewhat better pressure volume curve does not justify this design. It was decided, therefore, that the diaphragm blower will be built.

3.0 DESIGN OF THE BLOWER

As indicated above, the decision to have the operator seated while driving pedals was made quite early in the program. Therefore a bicycle was purchased to supply the parts of the drive: seat, frame, grips, pedals and a speed changer (built in the hub of the rear wheel of the bicycle). The speedchanger is of the planetary gear-type design, operated by a Bowden cable from the grips. When in direct drive, the wheel turns the fastest (as fast as the sprocket). The other two speeds available were 80% and 57.2% of the direct speed.

The conventional ratio of the bicycle pedal sprocket and rear wheel sprocket is 42 teeth vs 14 teeth. A 72-tooth sprocket and a 17-tooth sprocket were also purchased. The constant 55 rpm input from the pedals thus could be altered on the wheel from 330 to 95. Design of the blower itself, as originally conceived, was shown already on Fig. 1.

The neoprene diaphragm is held in place by the upper and lower heads, thus separating the two halves of the double acting pump. Movement of the diaphragm is done by the "piston rod" and retainer. The rod is driven by a Scotch yoke and guided by two linear ball bearings. The Scotch yoke is driven from the flywheel by a roller mounted on a ball bearing. The flywheel accommodates the speedchanger gear of the bicycle in a special hub.

The first valve design is shown on Fig. 5: a 4-in. diameter spring-loaded poppet valve.

The change on this original set-up necessitated by the findings of the first test series will be explained in detail later on.

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FIG 3. CIRCULAR VALVE

DRILL K (0.281)
4 HOLES EQUALLY
SPACED AS SHOWN

5.375 B.C.

$\frac{1}{2}$ DIA.

45°

SOFT RUBBER-LIKE MATERIAL
$\frac{1}{16}$ TH. x $\frac{1}{4}$ WIDE

$4\frac{1}{4}$ DIA.
ALUM.

$\frac{3}{8}$

$\frac{3}{8}$

PRESS
FIT

PRESS
FIT

SUIT.
SPRING

$\frac{1}{4}$ DIA.

TEFLON BUSH.

$\frac{1}{8}$

$\frac{1}{8}$

$\frac{11}{32}$

$\frac{3}{32}$ R

$\frac{3}{8}$

$\frac{1}{32}$

$\frac{5}{8}$ DIA.

$\frac{9}{64}$ DIA.

$\frac{5}{8}$ DIA.

4 DIA.

$4\frac{1}{8}$ DIA.

6 DIA.

$\frac{3}{32}$

$\frac{3}{16}$

$\frac{1}{16}$

ST. STL.

$\frac{1}{4}$-20 x $1\frac{1}{2}$ LG (2 REQ.)

$\frac{1}{8}$ DIA. x $1\frac{7}{16}$ LG
ST. STL. ROD
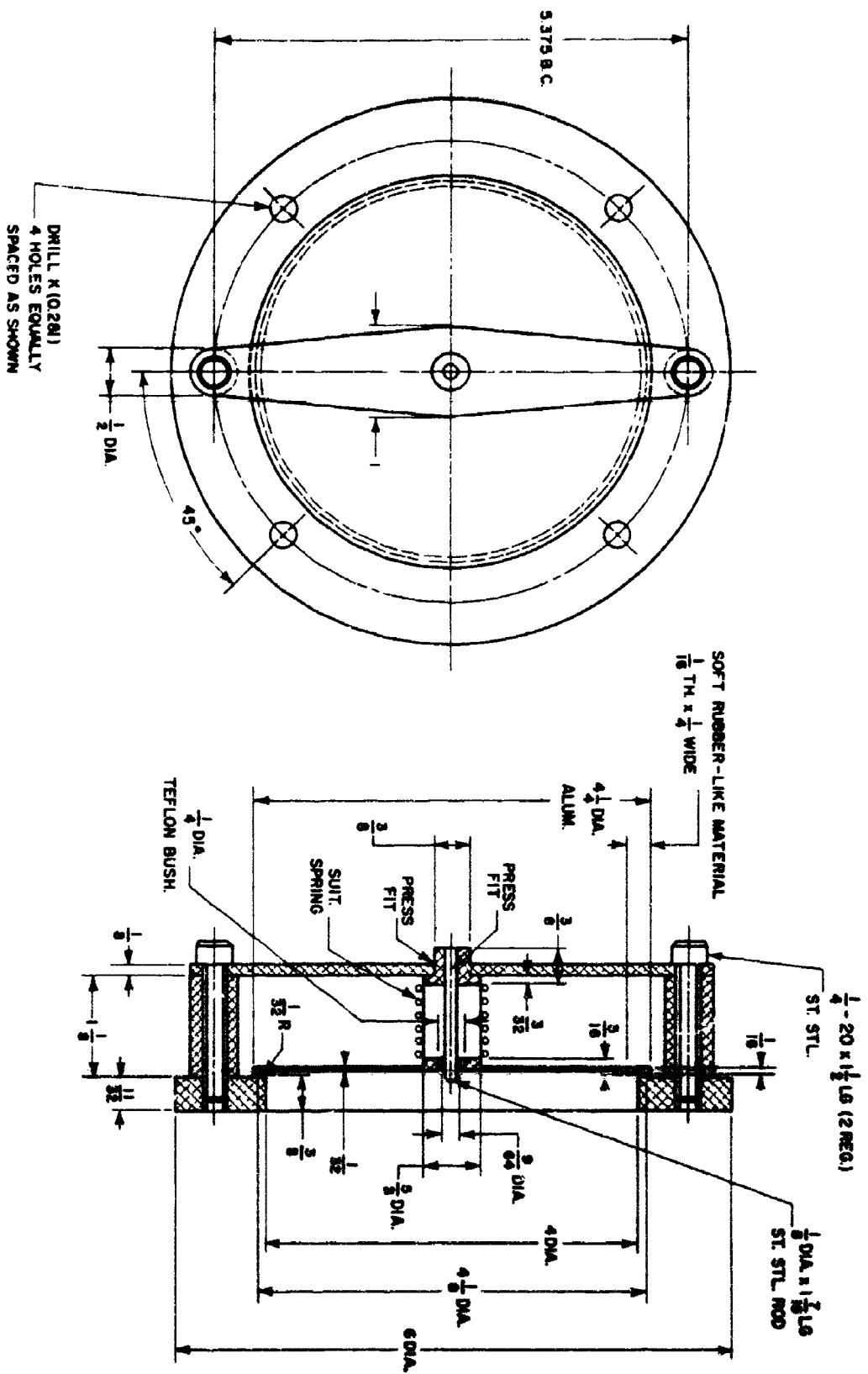
4.0 TEST SETUP

The power-linkage between the human operator and the blower is the chain drive between the pedal-sprocket and the blower sprocket. Therefore, as far as the blower is concerned, a measured power input from an electric motor at fixed speed adequately simulates the human operator provided the human operator pedals at an even rate. It was decided, therefore, that to overcome the difficulties inherent in measured human output, the blower will be developed and calibrated first, and when this work is done, the calibrated blower itself can be used as an instrument to evaluate the performance of different operators. Accordingly, the following test setup was assembled to test the blower (Fig. 6). Direct current electric motor (1) supplies power to 1:50* gearbox (2); the gearbox drives through the sprocket on its output shaft (3); chain-transmission (4); the air delivery of the blower is measured in a positive displacement "Roots Counter" (5); the number of turns of the counter is in direct proportion to the volume of air passed through the meter. (0.514 ft^3/rev. is the constant of the meter used). The delivery rate was determined by applying a timing device (6) which operated the Veeder-counter (7) for 0.514 minutes. Thus the revolutions of the meter counted on the Veeder-counter gave the cubic feet of air passed through the meter within a minute.

The blower discharged through its exhaust valves (8) directly into the room, the vacuum being generated upstream of the machine in the intake duct (9). This arrangement corresponds to a usual system configuration for certain shelter types, in which apparatus having a rather high aggregate resistance to air flow (such as filters, blast valves and heat transfer coils) may be provided on the inlet side of the blower.

Accordingly, the vacuum measured in the intake duct (9) between blower and meter constituted the pressure-differential the blower had to overcome. The pressure drop to create the vacuum in the duct was caused by placing fabric on a grid (10) in the intake of the meter or by a simple throttle in the duct between the meter and the blower (11).

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*in later tests a 1:25 gearbox was used
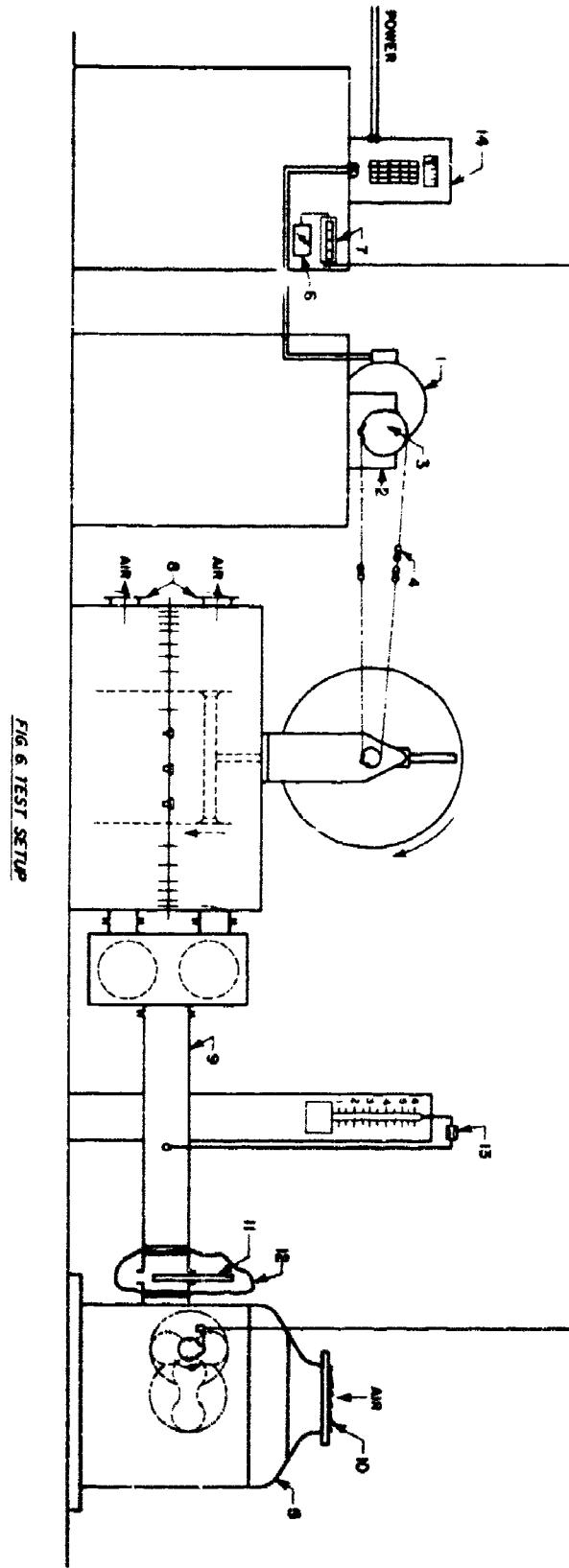
FIG. 6. TEST SETUP

5.0 INSTRUMENTATION

The cyclic delivery rate of the blower made it necessary that some consideration should be given to the instrumentation. As hydrodynamic type meters (orifices, Venturi meter, Pitot-tube, etc.) are influenced by velocity fluctuations, it was decided that a positive displacement meter should be used. The most readily available meter was the Roots meter mentioned above (courtesy of the Philadelphia Gas Company). This type of meter, however, gave another problem. The inertia of the rotating components of this large meter caused it to act as a pump on its own when the blower was at its dead-end location. This difficulty was eliminated by a flexible plastic receiver between blower and meter. The volume of air "delivered" by the meter while the blower stroke ended has been stored in the plastic bag (Fig. 6 (12)).

The fluctuating vacuum in the intake duct[10] was sensed through a one-legged manometer connected to the duct by a hose containing a capillary[13]. Thus the indicated vacuum was the time average of the pressure.

As the efficiency of the 1/8 HP, 2500 rpm (nominal) electric motor and the 1/8 HP, 1800 rpm (input) gear box was not known, the power input assembly (motor + flexible coupling + gearbox) was calibrated with a Prony-brake arrangement (Fig. 7), the power being registered electrically on a wattmeter[14], (Fig. 6), with an arbitrary scale. Calibration curves are shown on Fig. 8.

The test set-up shown on Fig. 6 was used during the entire program, with one major change. As it will be seen in the next section, the diaphragm blower did not prove satisfactory and the blower was used as a test vehicle to develop components rather than a final design. During the last part of the test program, the circular, 2-ft. diameter base of the blower (see Fig. 1) was substituted with a 26" x 26" x 36", rectangular body.
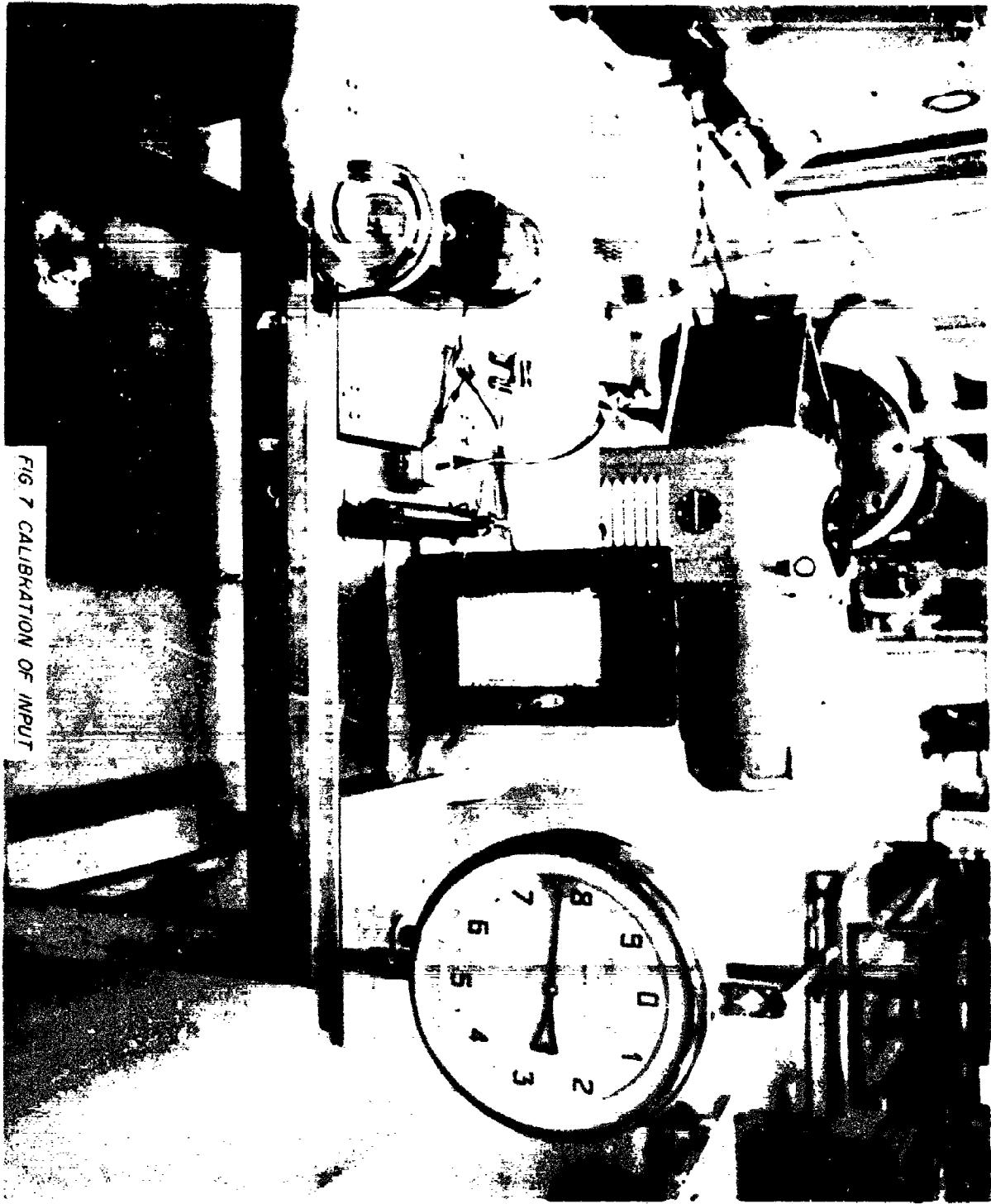
FIG 7 CALIBRATION OF INPUT

INPUT HORSEPOWER

WATTMETER READING

Δ-----Δ   60 RPM AT GEARBOX
X-----X   80 RPM AT GEARBOX
O——O   115 RPM AT GEARBOX
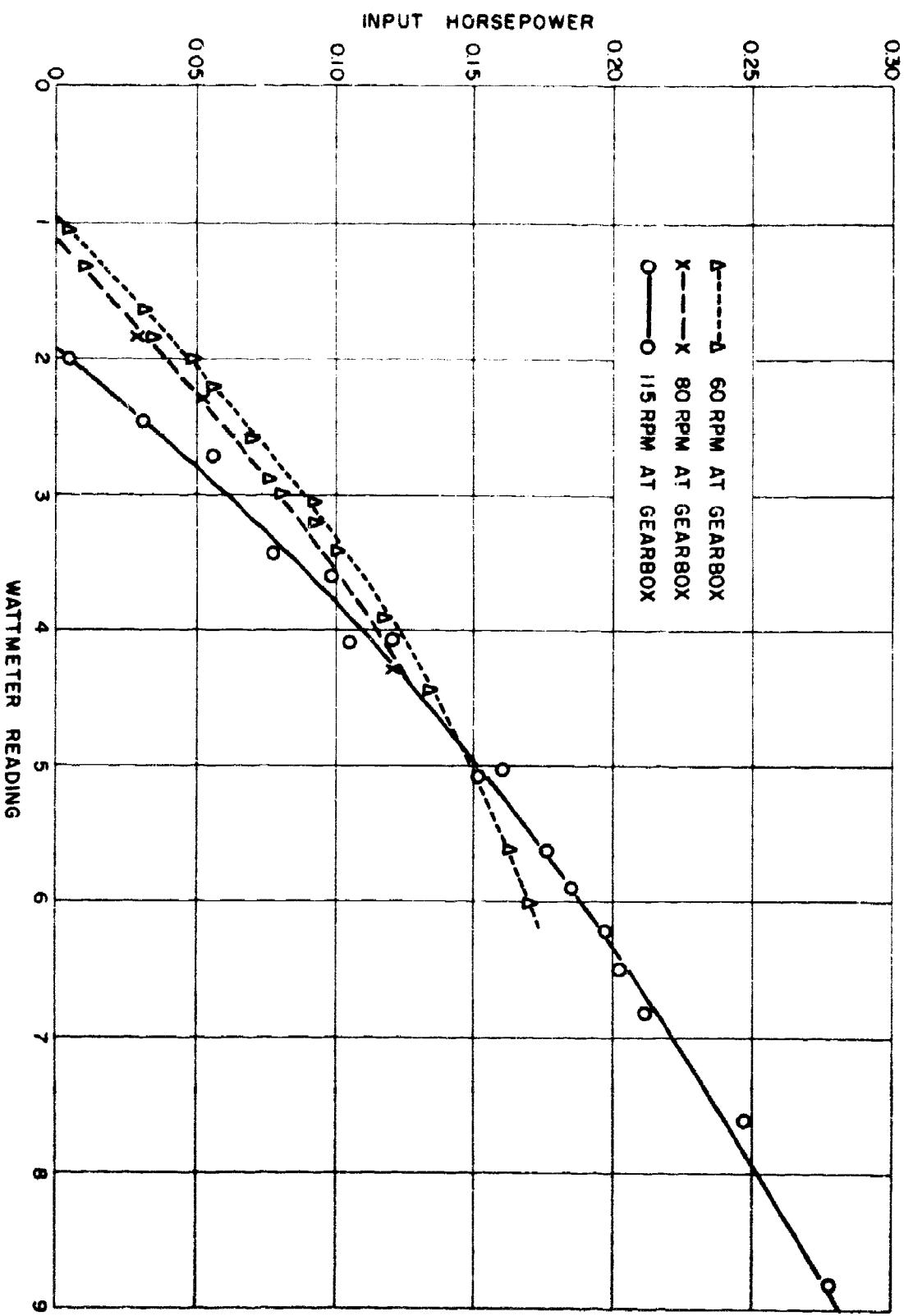
FIG 8. CALIBRATION CURVES OF 1:25 GEARBOX AND MOTOR

This box contained the 1-ft. cylinder and the rather parts of the original blower (Scotch yoke, piston rod, fly-wheel, bearings, drive), but it was easier to disassemble than the original machine: windows permit the observation of valves inside (Fig. 14). The large dimensions of the box, however, increased the clearance volume, as will be discussed later.

6.0 DEVELOPMENT WORK

6.1 PISTON DEVELOPMENT

6.1.1 First Tests with Diaphragm Blower

When the diaphragm blower (Fig. 1) was first tested, the blower did not display at all the characteristics of a positive displacement machine. Volumetric efficiency was 0.75 (max.) at no vacuum. When the intake was throttled and a modest 1 3/4-in. w. vacuum obtained, the volumetric efficiency dropped to 3.8%. Changing the valves improved the volumetric efficiency, but not above 41% at 2.0 in. w. vacuum. Visual inspection through one of the exhaust valves when the machine was operating revealed that the displacement (i.e. when the diaphragm assumes a cone-shape) takes place only at such time when the valves are opening or not quite closed yet, around the dead-end positions of the piston. In other words, during the most part of the stroke the diaphragm is ineffective. When it becomes effective, the valves are not. There were no simple means to overcome this difficulty (inherent in the non-rigid diaphragm) with valves that opened or closed under the impact of the flow through them, therefore the idea of a simple diaphragm had to be abandoned.

6.1.2 Second Test Series - Bellofram Diaphragm

The remedy to the above described situation would be a substantially rigid element: a piston. As the displacement of a piston is a cylinder rather than a cone (as in the case of a diaphragm) the same

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volume requires a smaller diameter. In order to utilize the exist-
ing set-up, a 12-in. diameter cylinder was placed coaxially inside
the original 24-in. diameter housing (Fig. 1). Two rolling diaphragms
(Bellofram Corporation) prevented the air from escaping through the
clearance between cylinder and piston from the pressure side of the
piston to the suction side (Fig. 9). It turned out, however, that

1) the Bellofram material does not have enough resilience
to keep its convex shape reliably against pressure
unless the toroidal air volume "A" between cylinder,
piston and the two Bellofram diaphragms (Fig. 9) was
held under constant vacuum from an auxiliary source
(the amount of vacuum in "A" has to be more than the
vacuum generated by the blower itself).

2) the force needed to ply the Bellofram was found to be
5.5 pounds (approximately). This force corresponds to
a power consumption equivalent to 25% of the useful
output (at 4.5-in. w. pressure). It was concluded
that it should be possible to make a piston with
sealing rings having less friction than the 5.5 pounds
of the Bellofram and, of course, without the need of
maintaining a vacuum around the piston.

6.1.3 Third Test Series - Conventional Piston with Felt Sealing

As it was difficult to anticipate the best trade-off between
friction losses and leakage (assuming that lower friction will result in
higher leakage loss) the first try was to obtain low friction with a con-
ventional piston and see what losses will occur. As a first try, two felt
rings were clamped between two masonite discs. The rings, bent into positions
parallel to the cylinder wall, formed two "cups", one in each direction.
This set-up had very low friction, but the leakage due to the piston could
not be kept below 22 cfm at 4.5 in. w. pressure differential. (The piston

STROKE 6

"A"

12.6"
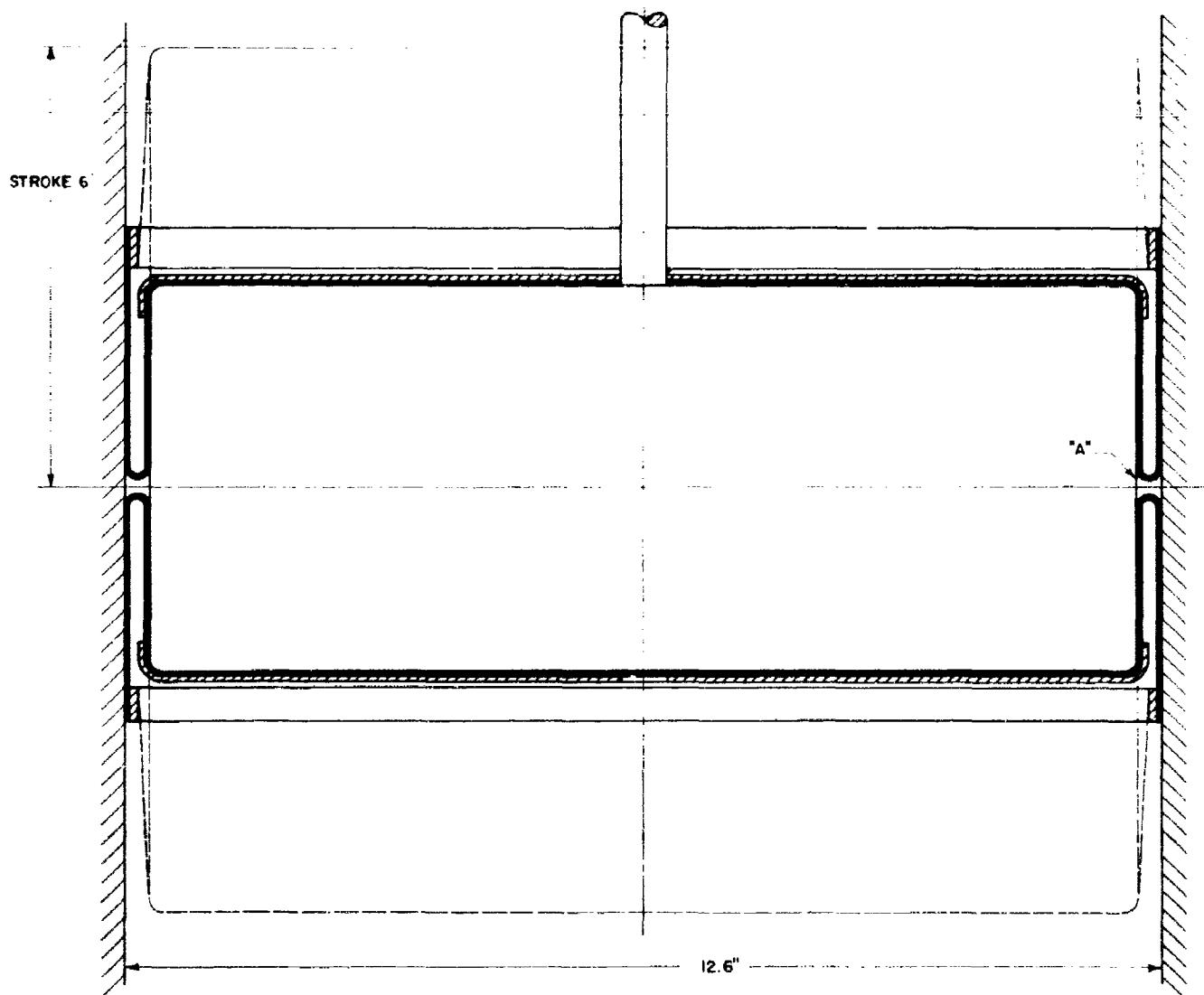
FIG. 9 ROLLING DIAPHRAGM

is not the only source of leakage; valves and the box also contribute to leakage.) This was considered too high and a better sealing piston was made.

6.1.4 Fourth Series - Rubber-Sealed Pistons

The first try at a rubber-sealed piston is shown on Fig. 10. It was intended to explore the gross influence of a few variables on sealing performance using the minimum amount of work for changes. In this preliminary investigation, the sealing rubber was originally a flat strip distorted into two cone shapes by applying the wire shown on the drawing. It was found that $\beta = 90^{\circ}$, A = 1/8" (approx.), t = 1/16" - 1/8" neoprene strip can achieve close to η_v = 90% volumetric efficiency. The friction on the cylinder wall, however, was rather high: 6-7 lb. force was needed to move the piston.

The next step, therefore, was to build a piston as shown on Fig. 11. The neoprene seal was double cone-shaped (distortion free outside the cylinder) and the force exerted on the cylinder wall could be closely controlled by the interference between cylinder diameter and cup diameter. It was found that 1/16" neoprene with 1/16" radial interference will cause less than 5 lb. friction force. A thin layer of grease was applied to the cylinder surface. The other dimensions of the seal are the ones shown on Fig. 10. This piston arrangement was used in the final tests. The volumetric loss due to this piston was less than 15% at 4.5" of w., 110 rpm (eventual valve-leakage losses were not separated; for "clearance volume" losses, see Section 7.0 and Appendix 2) During the final tests, the friction force was only 1.6 lb.

When the blower was operated at 6.0" w. vacuum, the leakage due to the piston changed only insignificantly. It can be stated, therefore, that the objective of building a blower insensitive to pressure has been achieved.
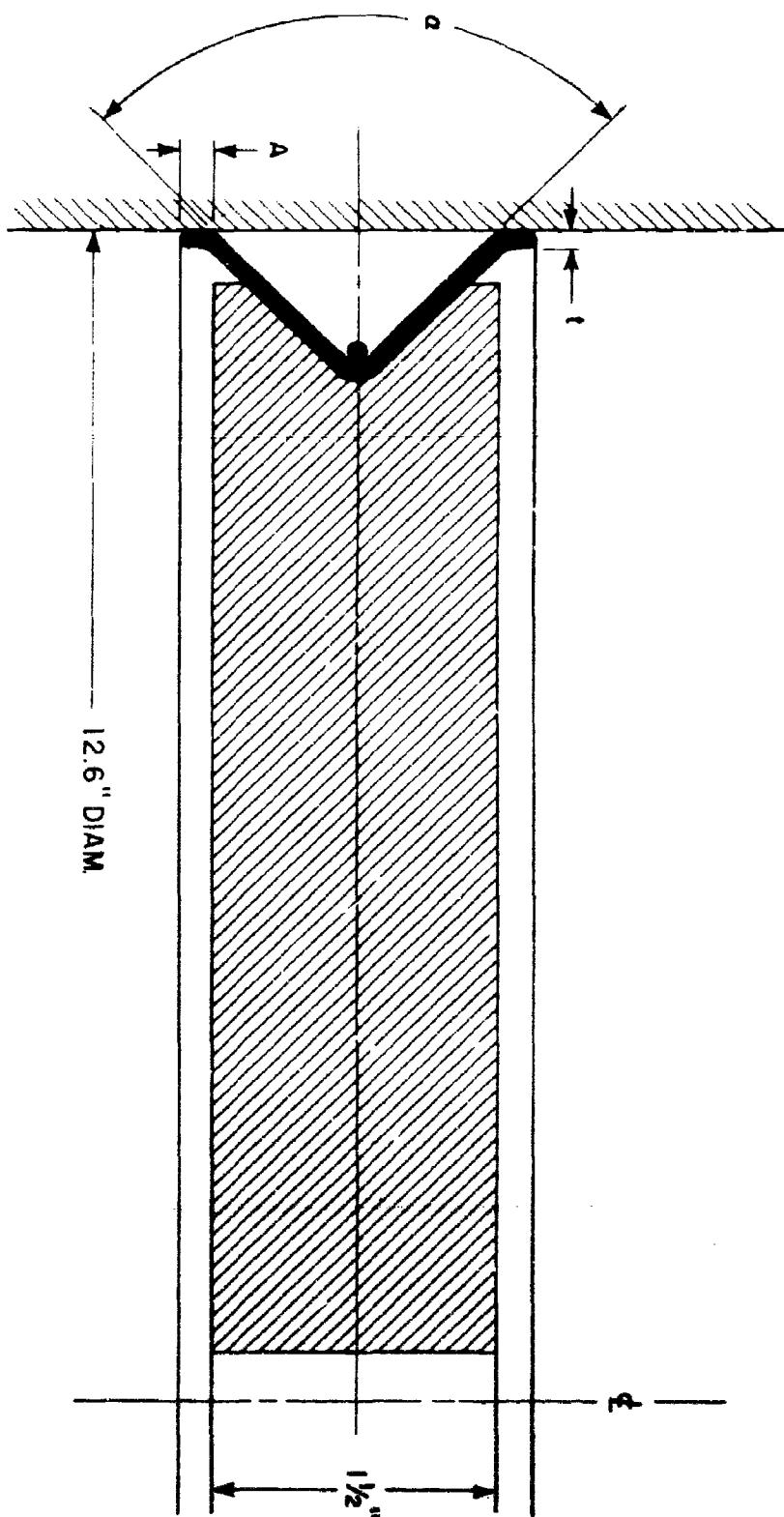
FIG. 10 FIRST RUBBER SEALED PISTON

12.6" DIAM.

1½"

$11\frac{15}{16}"$ DIAM.

$\frac{1}{2}"$ DIAM.

$\frac{3}{4}"$

45°

$\frac{1}{16}"$

① Wood Piston
1 Piece

$\frac{15}{16}"$

$r = 8\frac{27}{32}"$

244°

$12\frac{5}{8}"$ DIAM.

$\frac{1}{8}"$

$\frac{15}{16}"$

45°

② Rubber Cup
2 Pieces

$\frac{5}{32}"$

A ———— A

$11\frac{5}{8}"$ DIAM.

45°

$\frac{5}{16}"$

$\frac{5}{32}$

$\frac{1}{16}"$

SECTION A-A

$\frac{5}{32}$
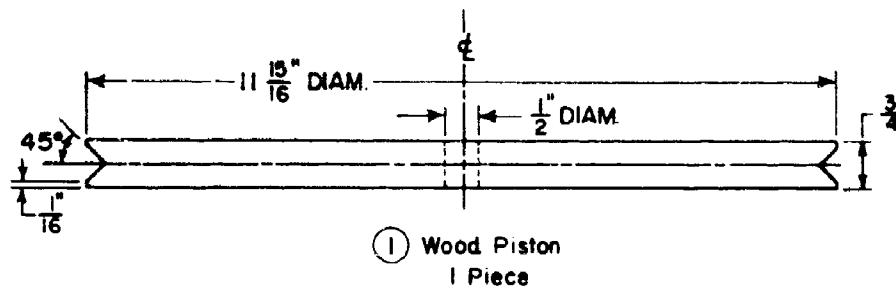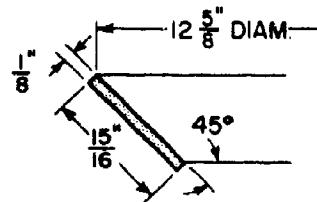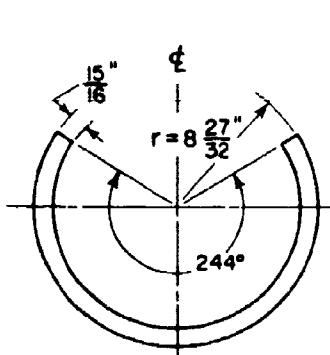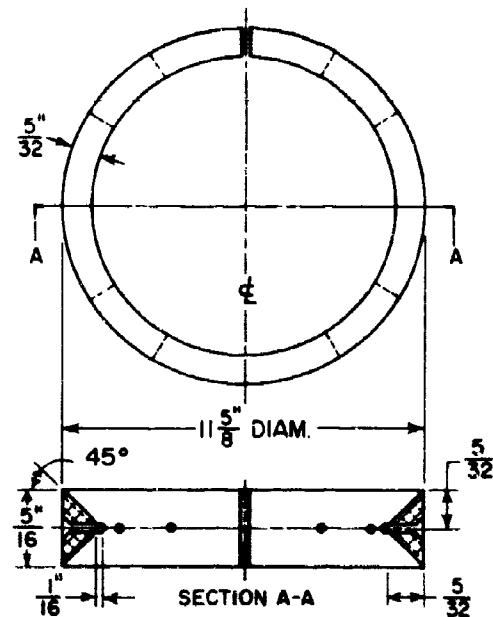
③ Split Aluminum Ring
1 Piece

②

①

③

②
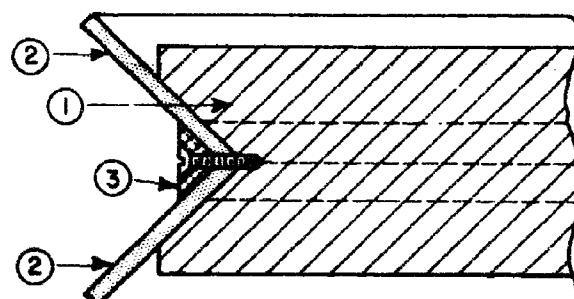
End Assembly

FIG. 11 FINAL PISTON ASSEMBLY

6.2 VALVE DEVELOPMENT

As it was described under "Design of the Blower" the machine was originally equipped with four poppet-type valves (Fig. 5). The valves, however, were not satisfactory. They were very noisy (though the noise level was considerably reduced applying sound-proofing material on the poppets and the seats as well). They did not open the same way all the time (to keep friction losses down only one very thin guidance rod was used and the poppets got stuck occasionally), but the greatest difficulty was the relatively high pressure drop necessary to operate them. In order to move the poppets fast, $k = 0.5$ lb/in. spring constant was necessary. The valve diameter being $4''$, at least $1''$ lift was needed to obtain the necessary discharge area. The resulting 0.5 lb. spring force in the open position had to come from aerodynamic forces of the flow. The valve area being $1/12$-ft.^{2} (approx.) the 0.5 lb. force corresponds to 6 lb/ft^2 pressure. As the rated pressure difference is 4.5 in. w. $= 23.4$ lb/ft^2, the power needed to keep the valves open is a sizable amount of the useful power of the blower.

Dr. C. H. Kearny proposed a valve design to the Office of Civil Defense (Fig. 12) for simple air-moving devices (like fans to maintain inner air movement in long, narrow shelters, etc.) and suggested trying it in the manual blower.

It can be seen from Fig. 12 that the response of this type of valve would be very fast as the mass of the moving parts is very small. At the same time the resistance of the valve can be kept very low as the grid is a small obstacle to the flow and no change in flow direction is involved.

Testing this "curtain-and-screen" valve, it was observed that the light plastic curtain was rather susceptible to wear at the rapid rate of impingement to the nylon grid. A curtain made of a heavier material, in turn, would not have closed as well as the light plastic. The basic

COLLAPSIBLE PLASTIC
CURTAIN

NYLON SCREEN

AIR
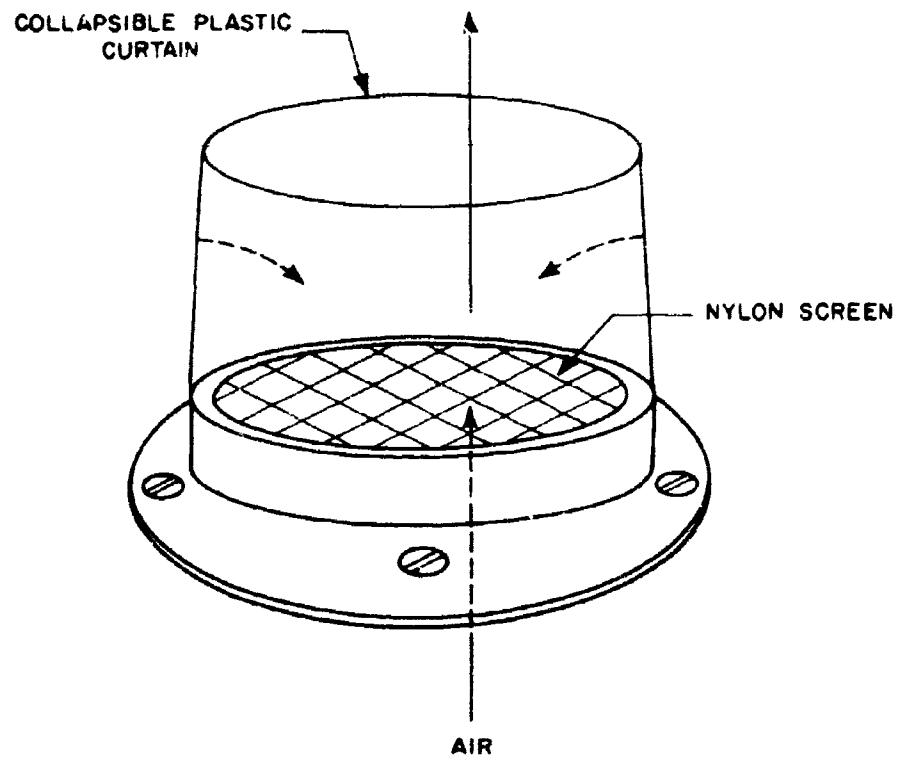
FIG. 12. CURTAIN AND SCREEN VALVE

(DR. C.H. KEARNY)

idea, however, seemed to be worthwhile to retain in a somewhat different set-up. It seemed attractive to have a light and pliable element open and close while the structural strength needed to brace the valve against the pressure difference in closed position could be delivered by another stationary element - a grid.

In order to obtain good sealing with the valve closed it was obvious that the heavier material had to be flat when closed. A rubber disc has been selected. The disc could be fastened in the inner area, either at the center (at one point) or around a concentric, small circle. The latter design was found to be better, probably because of leakage through the radial cuts in case of the other design. The thickness of the neoprene disc and the radius of the inner circle was found after some experimentation. As the material is more resistant to wear than the plastic curtain of Fig. 12, a metal grid could be used (Fig. 13).

The original 4" diameter valves of the blower were selected on the assumption that the poppet design will be satisfactory and a 1" lift (1/4 of the diameter) will cause only moderate pressure losses (0.1 - 0.2 in. of w. out of Δp = 4.5 in., at 150 cfm). The grid valve, however, does not open its total area, not even its entire circumference (at one diameter it always remains closed). The effective open are, therefore, is much smaller than in the case of a poppet valve. It was found that valve losses can be as high as 1 in. w. at half the rated flow rate. The obvious way of improvement is, of course, the increase of the valve diameter.

It was found that enlarging one exhaust valve from 4" diameter to 6" diameter, the efficiency of the blower increased by 6%. Therefore, when the round, original casing was replaced by the large cube-shaped housing (Fig. 14, see section 5.0) all four valves were enlarged to 6" diameter.

6.3 SURGE TANK DEVELOPMENT

The best efficiency obtained with the above described changes and improvements was only around 40% (depending somewhat on pressure and speed). Because of the low efficiency, it was decided that in addition to the efficiency
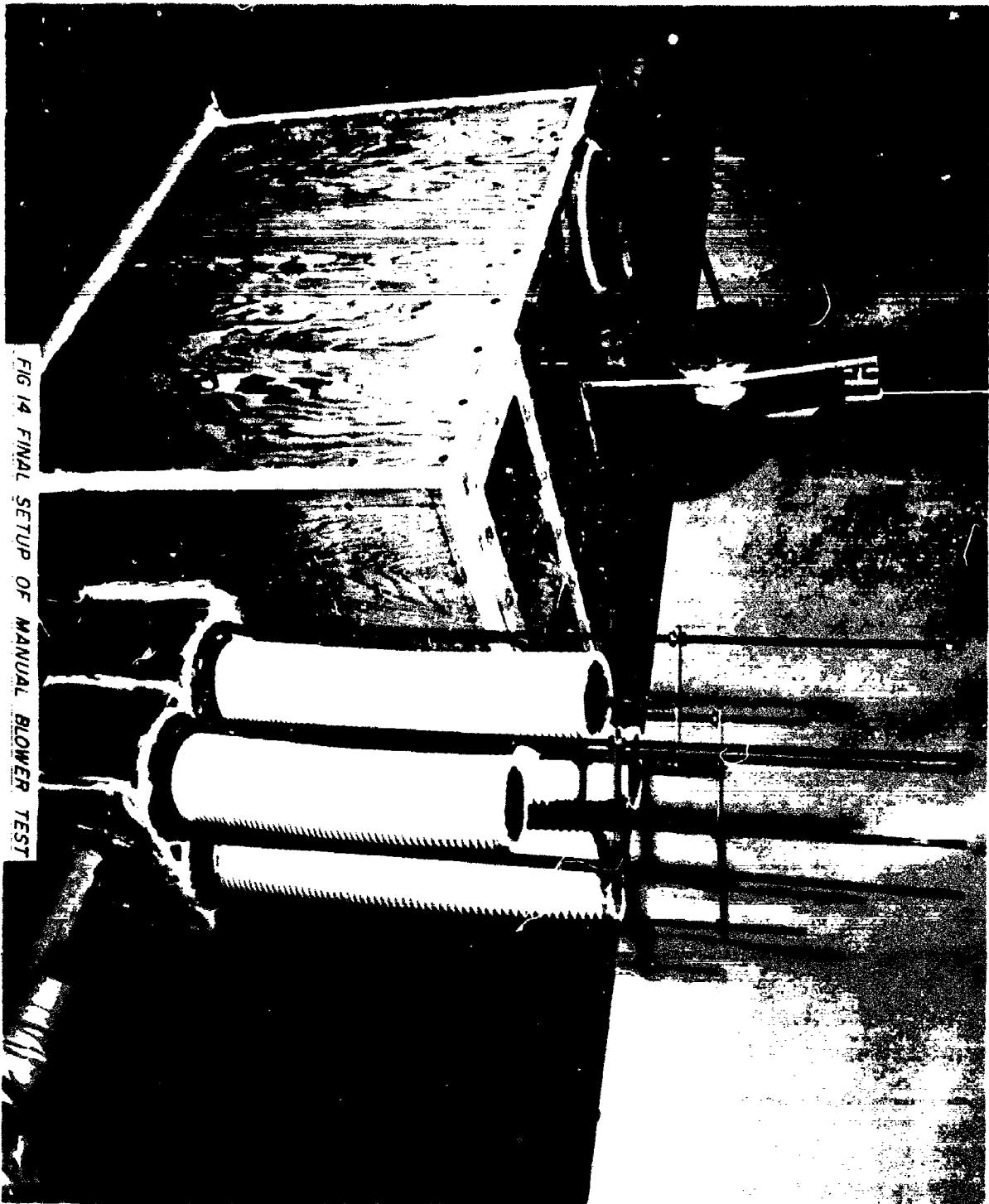
FIG 14 FINAL SETUP OF MANUAL BLOWER TEST

FIG 13 GRID VALVE

COVER

COVER

COVER

GRID

AIR

$\frac{7}{8}$ DIA.

$\frac{1}{32}$
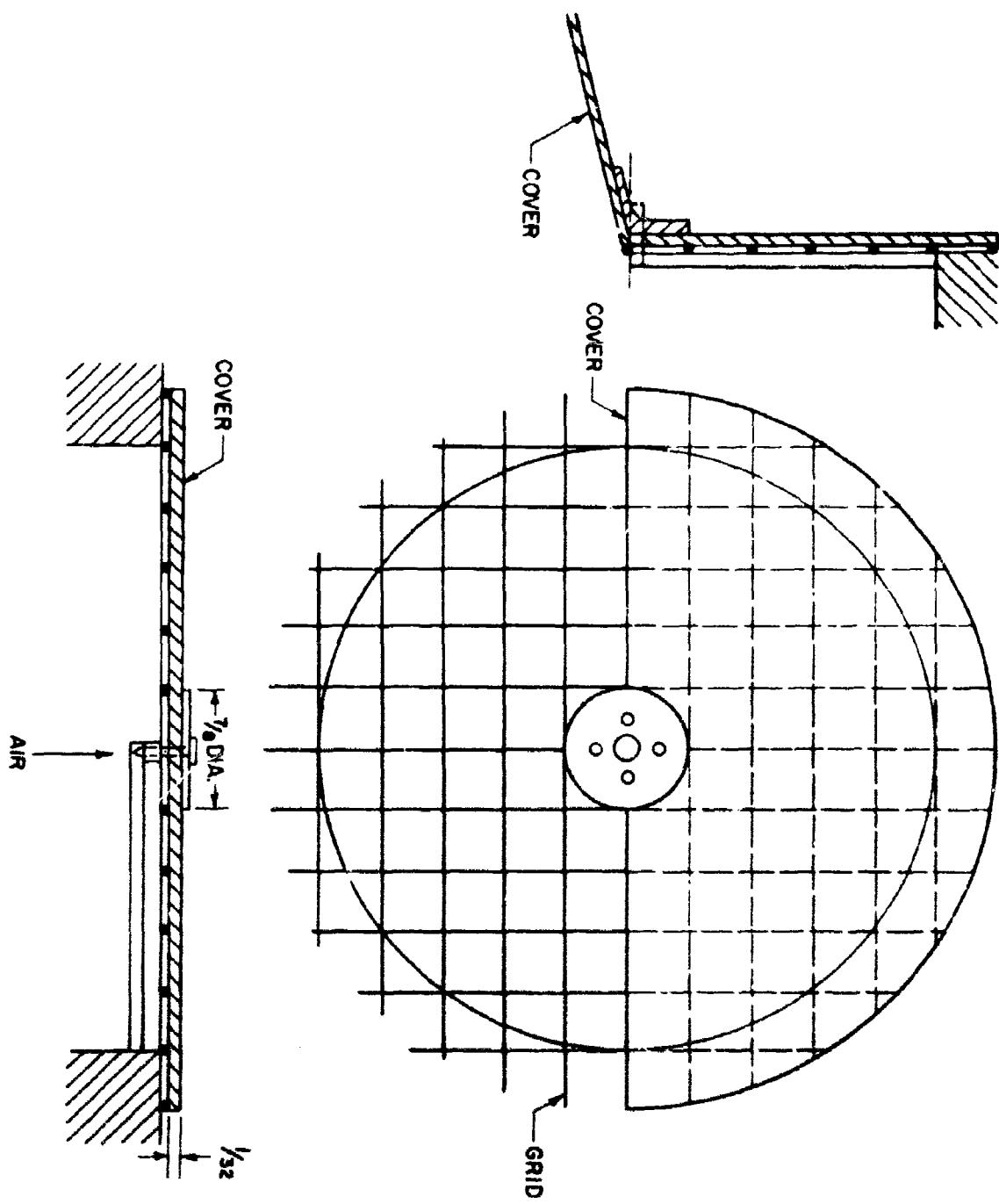
improvement due to the valve enlargement another source of losses will be attacked, namely, the additional losses due to the unsteady flow inherent to positive displacement machines.

The source of losses can be explained qualitatively as follows: The flow rate in the blower, because of the Scotch yoke, is basically a sinusoidal function of time. The pressure drop, therefore, follows a $\sin^2 \omega t$ function, in case of a quadratic resistance. The useful output of the blower is the time-average of the sinusoidal delivery-rate, multiplied by the minimum amount of pressure to produce it ($\sin^2 \omega t$). The power required to maintain this output, however, will be proportional to the average of the product of instantaneous delivery and pressure, i.e. $\sin^3 \omega t$. As it is well known, the average of $\sin^3 \omega t$ is more than the product of the average of $\sin^2 \omega t$ and $\sin \omega t$. Therefore, more power is needed to drive a blower with an intermittent delivery rate than it is necessary for a constant delivery machine. It can be shown that the ratio of power needed in the two cases is $\pi^2 : 6$ (Appendix 2). It should be noted that the $\pi^2 : 6$ ratio refers to the useful power only. A method to reduce this excess power can be outlined as follows.

A chamber has to be installed into the conduit between blower and flow resistance. One wall of this chamber is elastic. When the instantaneous pressure is equal to the average pressure (4.5 in. w. in our case) the chamber wall is in equilibrium (its spring constant must be chosen appropriately). When the vacuum increases (the piston accelerates) the volume of the chamber decreases (the elastic section deflects under the outer atmospheric pressure). Therefore, the flow rate at the flow resistance does not have to increase so much, as the chamber provides a certain amount of air needed in the blower. When the piston decelerates, the vacuum decreases, the elastic wall increases the chamber volume and more air flows through the filter than the blower calls for. The excess air will be stored in the chamber.

It can be shown (Appendix 2) that if Q_p is the volume of
the pump (stroke x piston area) the necessary "surge tank" volume
will be

$$v = \frac{0.33}{\pi} Q_p$$

It is obvious that the pressure fluctuation (and therefore the excess
power required) cannot completely be eliminated as the operation of the
storage tank is based on a certain amount of pressure fluctuation. It
can be shown (Appendix 2) that if $\Delta\eta$ is the loss in efficiency,

$$\Delta\eta = \left(\frac{\Delta p_s}{2\Delta p_{av}}\right)^2$$

where Δp_s is the pressure fluctuation in the surge tank and Δp_{av} is the
average pressure level. In case of $\Delta p_{av} = 4.5$ in. w., $\Delta p_s = \pm 1$ in. w.
requires 1.23% excess power, which is quite small compared with the 64%
calculated from the π^2:6 ratio referred to above. The 12.6-in. piston
diameter and 7-in. stroke result in a blower volume of 870 in^3. The stored
volume will be, therefore, 91 in^3, approximately. After a short investi-
gation it was found that the simplest way to make a surge tank would be
the use of commercially available flexible hose material. The only avail-
able hose had a nominal diameter of 4 inches. The hose has a steel spring
core, the material is plastic, cast on the spring. Four hoses were used,
each was to have a displacement of 23 in^3 (approx.). The 4-in. diameter
prescribes a 1.8 in. = $\pm$ 0.9-in. movement of the hoses; the $\pm$ .9 in movement
and the $\pm$ 1 in pressure fluctuation should determine the spring constant, if
$\pm$ 1 in. w. pressure fluctuation is assumed. The spring constant of the hose,
of course, could not be changed, except that it could be made somewhat stiffer.
There was no opportunity for much experimentation with the hoses (the end of
the project was very close by the time the surge tank development got under-
way), and the final tests were made with surge tanks which did not give the
best possible results. The measured spring constant of the four hoses was
0.8 ± 0.1 in/lb. The measurement of the spring constant was made as the
hoses were used in the final tests (December 14-15, 1964).

7.0 RESULTS

The program ended with test runs on the 14th and 15th of
December, 1964, using the components described in the previous sections.
The following results were obtained:

TABLE I

No.	RPM	Input Power HP	Flow CFM	Pressure in. w.	Output HP	Effectiveness[1] or Efficiency %
1	110	0.088	86	4.45	0.0596	68.0
2	110	0.102	81	5.75	0.0725	71.0
3	115	0.033	109	0.35	0.0059	18.0
4	110	0.049	97	1.80	0.0272	55.5
5	109.5	0.056	94.5	2.25	0.033	59.5
6	110	0.0875	85	4.6	0.0607	69.5
7	110	0.086	86	4.6	0.0615	71.7
8	110	0.099	82	5.6	0.0715	72.0
9	145	0.043	139	0.5	0.0109	25.0
10	145	0.126	112	4.45	0.0775	61.5
11	145	0.0875	124	2.85	0.0552	63.0
12	145	0.1475	107	5.65	0.094	63.5
13	145	0.121	113	4.55	0.080	66.5
14	185	0.093	165	0.6	0.0154	16.6
15	285	0.22	220	1.0	0.0343	15.6

The entries in Table I were obtained either directly from instrument
readings (RPM, flow (V), pressure (Δp)) or indirectly (input power (P) from
wattmeter readings using the calibration curves of Fig. 8). The entries for
output (P_{op}) and effectiveness (η) were calculated:

$$P_{op} = \frac{V\Delta p}{6410}$$

and

$$\eta = \frac{P_{op}}{P}$$

The units in the above formulae were the ones indicated in Table I. Results
for 110 and 145 rpm are plotted on Fig. 15.


[*] in certain tests same as "Efficiency", see Appendix 3
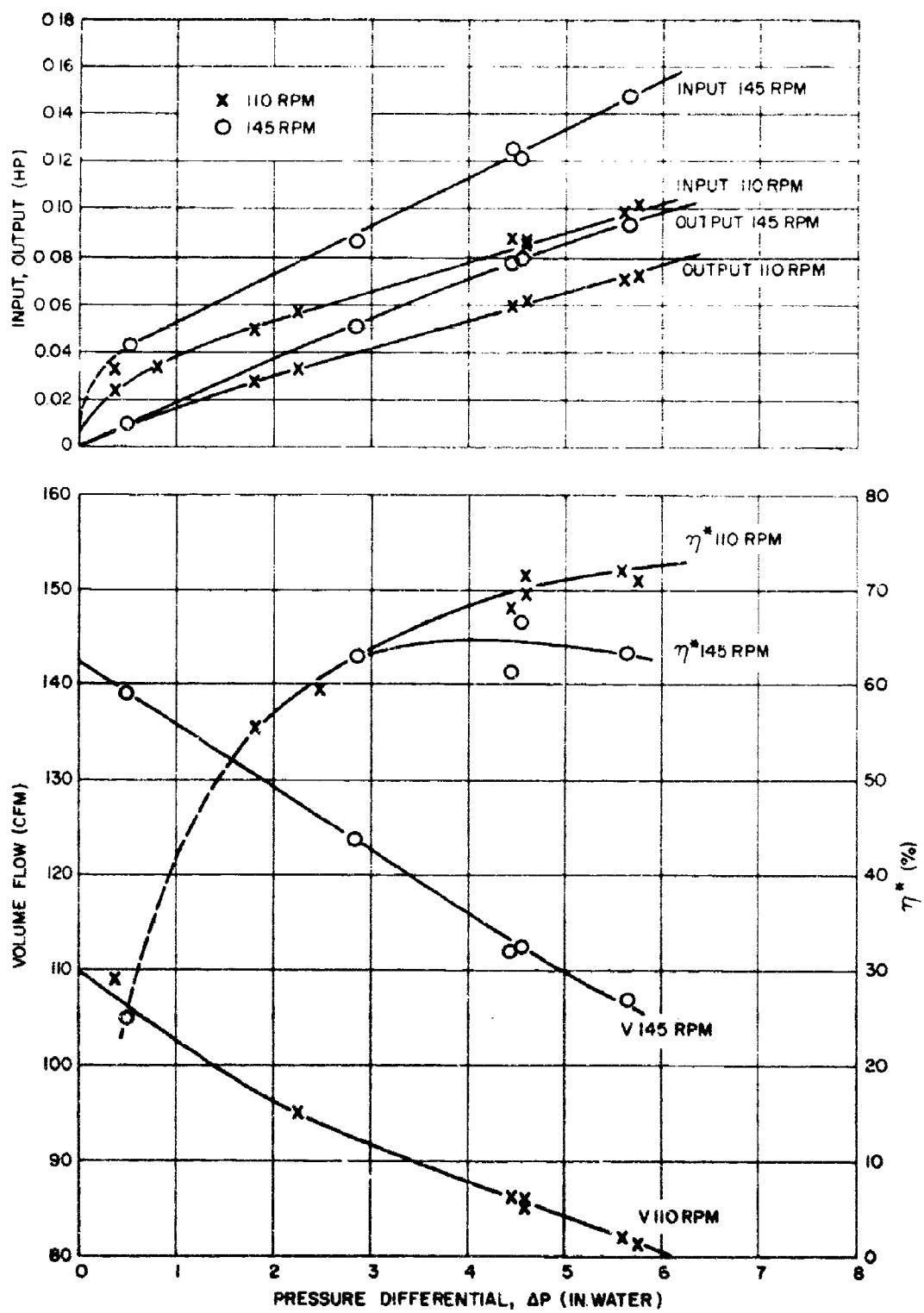
FIG. 15. BLOWER PERFORMANCE WITH SURGE TANK

(DATA FROM TABLE I)

As the displacement of the 12.6-in. diameter, 7-in. stroke piston is exactly 1 ft^3, the numerical value of the rpm (n) is equal to the theoretical delivery rate of the blower. The volumetric efficiency, therefore, can be calculated from the formula

$$\eta_v = \frac{V}{n}$$

It can be seen from Table I that test results could be repeated with an accuracy of approximately $\pm$ 2%. The efficiency of the blower drops several percentage points when higher speeds are used. To find the reasons for the decrease in efficiency an attempt was made to calculate the losses. To separate the individual losses as much as possible the blower was tested, the surge tanks being immobilized. Results are shown in Table II and Fig. 16.

TABLE II

No.	RPM	Input Power HP	Flow CFM	Pressure in. w.	Output HP	Effectiveness %
1	110	0.1020	85	5.0	0.063	65.0
2	110	0.0600	100	1.95	0.030	50.0
3	110	0.115	81	6.05	0.0765	66.6
4	110	0.0975	87	4.7	0.0635	65.0
5	110	0.095	87	4.65	0.0630	66.5
6	145	0.0975	130	2.05	0.0415	42.5
7	145	0.157	120	4.60	0.086	54.8
8	145	0.188	113	6.05	0.107	57.0
9	145	0.0585	132	0.55	0.011	18.8

As it is shown in Appendix 3, at the lower speed (110 rpm) the improvement in efficiency due to the application of the surge tanks is only one half of the theoretically obtainable maximum (approximately). Valve losses,

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*defined in Appendix 2, Eq. 13
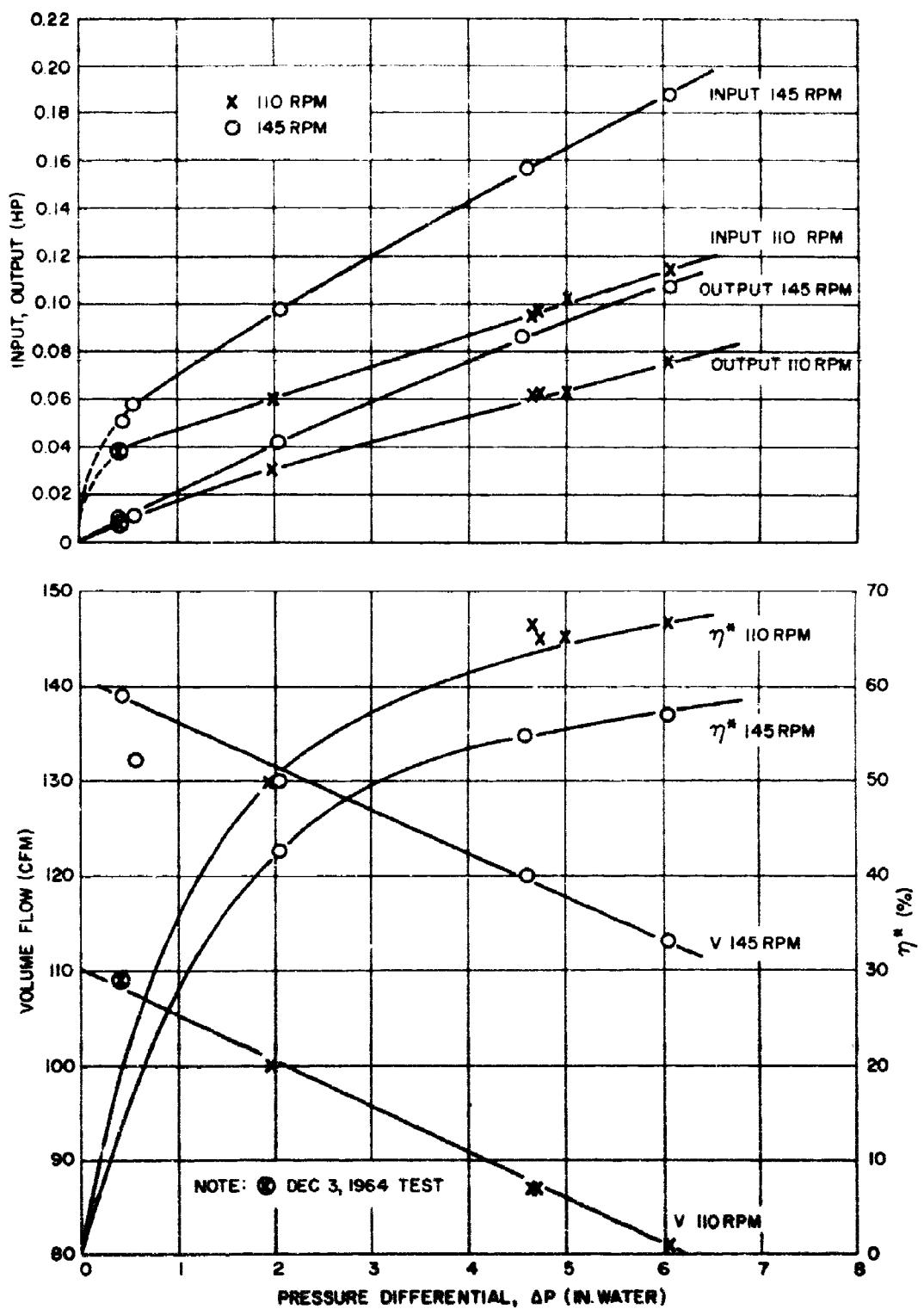
FIG. 16. BLOWER PERFORMANCE WITHOUT SURGE TANK
(DATA FROM TABLE II)

however, are so low that their amount cannot be shown within the accuracy of the test.

As is shown in Appendix 3, the decrease in losses due to tne application of the surge tank is not enough to consider the effectiveness for the 110 rpm runs at 4.5 in. w. as "efficiency".

For the 145 rpm run, however, the calculated "effectiveness" is really the conventional blower efficiency for $\Delta p = 4.5$ in. w. as the Δp_s disappears. (Unfortunately, the valve losses were rather high and this kept the total efficiency below 70% for 145 rpm). Blower losses are plotted in Fig. 17.

Searching for the reason behind the different performance of the surge tanks, it has been found that movement of the surge tanks was approximately 25% more for the higher speed run. A closer inspection revealed the fact that the volume provided by the surge tanks should not have been calculated with the nominal diameter, but with a smaller effective diameter. Consequently, the calculated 1.7 in. movement was not enough to provide the calculated 91 in^3 tank volume, more movement was needed.

8.0 DISCUSSION OF RESULTS

The results of the experiments show that the good efficiency and pressure holding capability of the blower is still not the best that could be obtained with such a machine.

In case of mass production, the friction between cylinder wall and piston can be further reduced by applying a teflon coating on the wall of the cylinder.

The delivery rate of the blower is only 112 cfm at the speed that gives the best efficiency (145 rpm). If 150 cfm should be obtained two measures could be taken: either drive the same blower faster or increase the piston
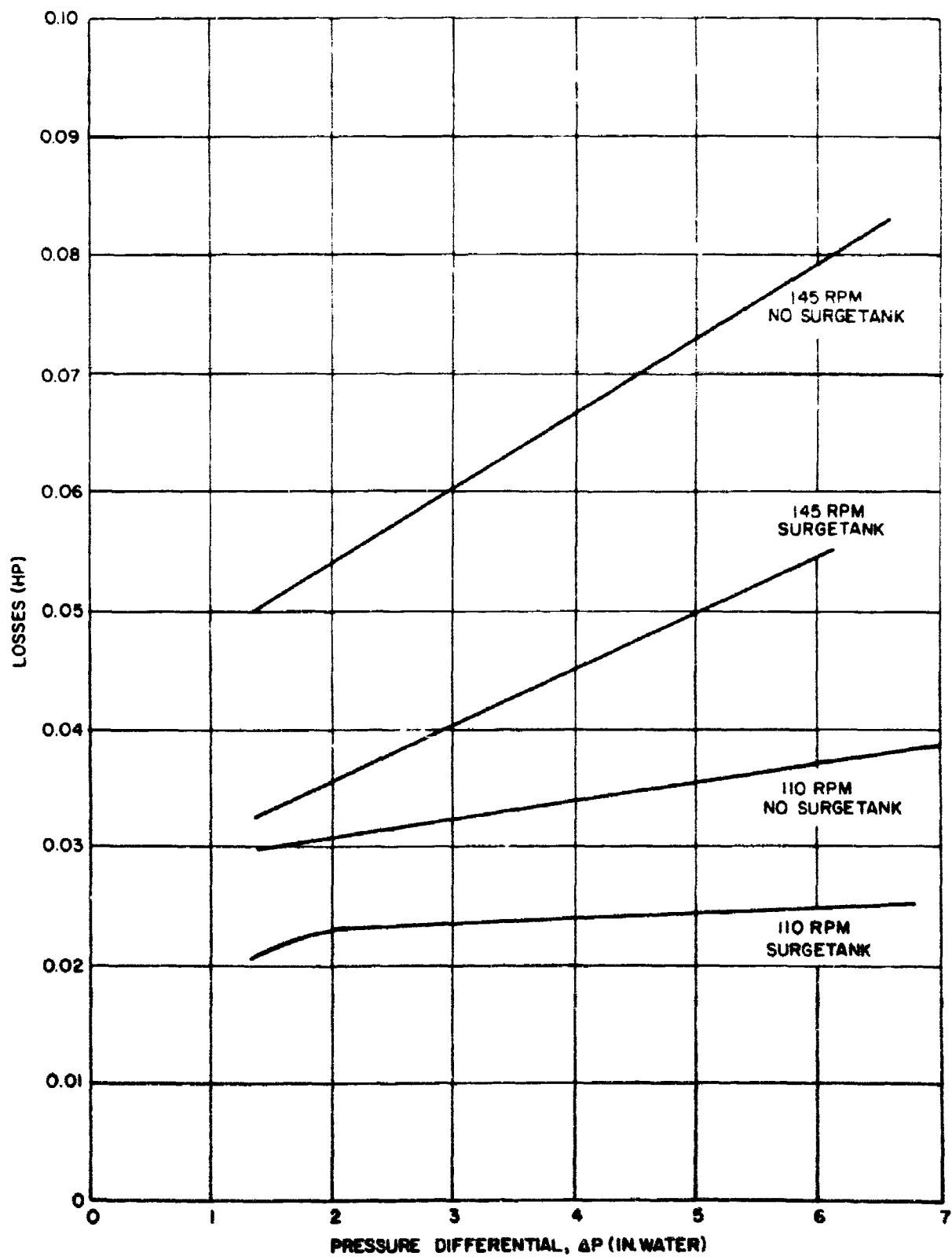
FIG. 17 BLOWER LOSSES

(and cylinder) diameter. It is conceivable that increasing the piston
diameter and keeping the speed the same, the efficiency of the blower will
increase; losses will not increase in proportion with the larger piston.
One has to keep the valve flow velocities the same, however. i.e.. the
valves have to be enlarged. An 18-in. diameter piston and 7-in. diameter
valves would be the upper limit. A combination of the two methods could
also be considered: slight increase in speed and piston diameter. Keeping
the speed the same is more advisable because of the dual purpose of the
machine. When 300 cfm has to be delivered (motor drive) the machine has to
be run at twice the speed of the manual operation. The lower the manual
speed the easier it is to design the machine mechanically for higher speeds.

The large clearance volume reduced the delivery rate and, in general,
made the machine bulky. There is no reason, however, that the clearance volume
should be as large as it was in this experimental machine. The inner flow
velocities in the machine should be kept at the level of the valve velocities.
The internal cross-sections, necessary to achieve valve velocities, are quite
small.

As it was pointed out before, the surge tanks were made of commercially
available material. In case of mass production, the material of the surge tanks
can be specially designed for the purpose. The general idea of the design used
during the tests seems to be attractive. A wire spiral gives strength and
elasticity to the tanks. A thin polymer sheet is cast around the spiral and
no additional fastening is needed between steel and wire. The spring constant
of the device can be prescribed beforehand if the material is designed for the
mass production of the blower.

9.0 RECOMMENDATIONS

The positive-displacement type of blower developed under this project has "high pressure-low volume" characteristics and could be used to best advantage in ventilating systems for shelters that have an objective requirement for good performance under such conditions. This type of air mover can operate efficiently over a wide range of conditions. It is apparent that high efficiencies in a positive-displacement blower are associated with low values of seal leakage rates, friction losses, clearance volume, air velocities through valves and approach ducts, and flow variability. Unbalanced dynamic forces as well as flow pulsation effects could be reduced by using multiple, double-acting cylinders displaced by a phase angle. Fans having "lower pressure-higher volume" characteristics would be quite satisfactory for ventilating identified fallout shelters in existing buildings, and designs for the positive-displacement type of blower could be adapted to meet the less rigorous requirements.

The outline of a blower, suited for mass production, is shown on Fig. 18. The sketch is based on the results of the tests as they were evaluated in the previous section.

As explained in Appendix 1, the bicycling position being preferred, the blower is shown equipped with a bicycle-type drive. The seat and the pedals, however, are detachable and the machine can be outfitted with hand drive also. Air inlet is on the bottom. The inlet duct may be connected to any one of three flanges; the blower, therefore, can face in three different directions. The surge tanks take up the two openings not used for the inlet pipe. As the inlet pipe will be of standard 6-in. diameter, the surge tanks should be interchangeable. The horizontal cylinder is 18-in. diameter; the sealing of the piston should be made in accordance with Fig. 10 and Section 6.1.4.

Valves have a 7-in. diameter. In other respects they are described on Fig. 13 and in Section 6.2.

EXHAUST VALVES

MOTOR

FLYWHEEL

SCOTCH
YOKE

INLET VALVE

28"

SURGE TANK

INLET PIPE

INLET VALVE

CYLINDER

PISTON ROD

PISTON

19"

DIVIDING
WALL
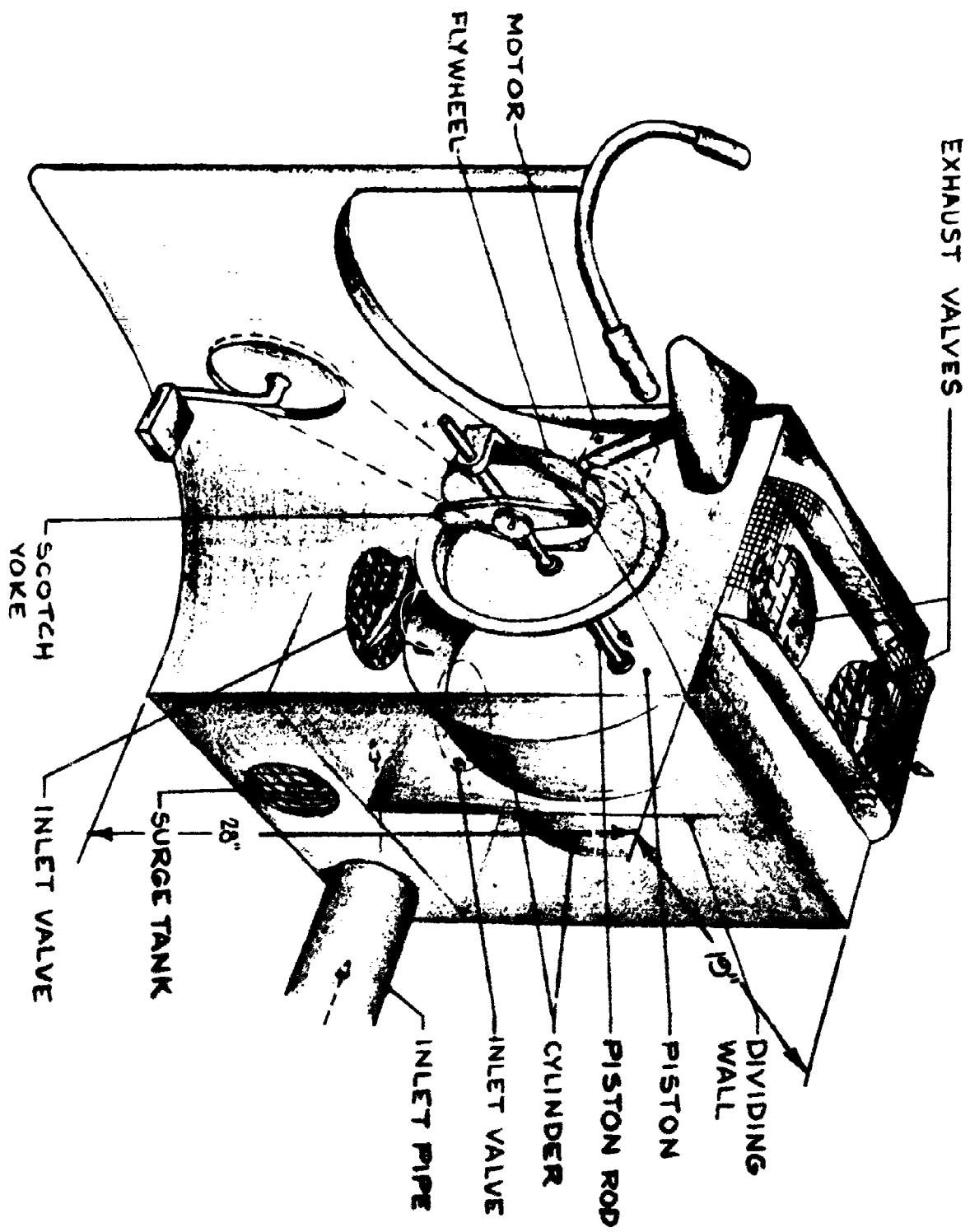
FIG. 18. MODIFIED DESIGN CONCEPT FOR MANUAL BLOWER

There was no determination of the size of the flywheel. It is recommended that a prototype of the machine should have provision to change the size of the flywheel, and tests should be made with different individuals to determine the best flywheel size.

The motor drives the pump through a frictional drive; it can be a standard A-C motor, no particular requirements have to be fulfilled. Disengagement of the motor will be done by lifting it away from the flywheel.

Guidance of the piston rod will be accomplished through straight bushings, similar to the ones used in the test equipment.

The blower has no preferred running direction. The height of the machine may be reduced by making the inlet chamber lower than the inlet pipe diameter (6 inches). In this case, the inlet pipe and the surge tanks would be attached to a 6" x 7" x 19" box alongside the blower. The height of the machine could be reduced to 24 inches (approx.).

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Approved by:

Nelson R. Droulard
Technical Director

Nomenclature

A: cross-sectional area, ft^2.

C: constant; unit defined in Equation 3, Appendix 2.

D, d: diameter, ft (Appendix 1).

D_s: specific diameter (Appendix 1), ft.

g: gravitational constant, ft/sec^2.

H: head of fluid, ft.

k: spring constant, lb/ft.

L: length, ft. (Appendix 1).

n: rotational speed, rpm (in Appendix 1 rps when so defined).

N_s: specific speed, see Appendix 1.

Δp: pressure differential, in. w.

P: power, HP

ΔP: power loss, HP

Q: volume of cylinder, ft^3 ;in Appendix 1: volumetric flow, ft^3/sec.

s: stored flow rate, ft^3/min.

S: stored quantity in surge tank, ft^3.

t: time, sec.

V: volumetric flow of air, ft^3/min.

W: power, unit consistent with other quantities used in Appendix 2.

Nomenclature (continued)

Greek letters

$\mathcal{L}, \beta, \gamma, \delta, \epsilon, \zeta, \eta, \Theta$: exponents used in Appendix 1, not used elsewhere.

ρ: density, lb/ft^3.

η: efficiency

η^*: effectiveness (definition in Appendix 2)

$\Delta\eta$: decrease in efficiency

Θ: intermediate variable (Appendix 2)

Π: dimensionless products (Appendix 1)

ω: frequency, rad/sec.

Subscripts

ad : adiabatic

av: average

c: clearance

cd: constant delivery rate

f: friction

h: hydraulic

in: input

L: leakage

m: mechanical

n: natural

o: maximum value of variable

op: output

p: pump (blower)

s: sinusoidal or surge tank or specific

t: total

th: theoretical

v: valve or volumetric

APPENDIX 1

SELECTION OF DRIVE AND TYPE OF BLOWER FOR A MANUALLY OPERATED SHELTER-BLOWER

A. INTRODUCTION

This Appendix summarizes the considerations leading to the selection of the drive and type of the manually operated shelter-blower, the development of which is the subject of Contract No. OCD-OS-62-280. (Franklin Institute Project No. 18G-B1983). Some of the material contained in Interim Report I of this program is repeated here for the sake of completeness and the convenience of the reader.

B. SELECTION OF THE DRIVE

Basically there are only two muscle-groups that the blower drive can be based upon: the hand-arm-shoulder group and the postural muscles of the legs and the back. The leg and back muscles differ from the other muscles in the body: they contain a higher percentage of muscle-hemoglobin than their non-postural equivalents. The relatively large amount of muscle-hemoglobin provides a reserve store of oxygen and also enables a rapid restoration of depleted oxygen. Consequently, the steady-state (i.e., several hours) work output of postural muscles is considerably greater than that of non-postural muscles; the difference becomes more marked as work time increases.(Ref. 6) The non-postural muscles are capable of more vigorous efforts but only for short periods of time (like the punches of a boxer).

Requirements of the blower-drive clearly indicate a design based on the "sustained effort" muscles; i.e., the back and leg muscles: a bicycle-type device. Experimental verification of the above reasoning can be

readily seen from Figure 1 and Figure 2 (both were adapted from Ref. 7). Figure 1 shows that less than 0.04 HP can be expected from a hand-cranking device beyond 15 minutes (1000 sec); best performance for 30 seconds - less than 0.07 HP. On the other hand, (Figure 2) pedalling power for 4 minutes (240 seconds) exceeds 0.5 HP; for 15 minutes, 0.4 HP; apparently 0.1 HP can be expected for many hours from average persons. (It would be half the output of an athlete, see dashed line of Figure 2.) There is obviously an order of magnitude difference between the performance of the two muscle groups in favor of the leg-back muscles.

C. CRANKING SPEED

Output of human power is obviously strongly influenced, among other factors, by the RPM at which this output is obtained. Ref. 2 does not give a definite answer to this problem, but it seems from previous work performed in the subject and reviewed on the reference that optimum speed is between 32 and 70 RPM and most likely around 52 RPM. For the purpose of designing the test equipment, it will be taken as 50 RPM. One can conclude, therefore, that for the purpose of the blower the human power available will be somewhat over 0.1 HP at 50 RPM.

D. SELECTION OF THE TYPE OF BLOWER

Basically, there are two ways to select the type of blower best suited for the application and the type of drive selected in the preceding section: one can consider all the available types (axial, centrifugal, various positive-displacement-type machines) on the market and discuss their various characteristics, making qualitative comparisons; or, find a quantity that could be used to represent the various types of blowers and find the optimum
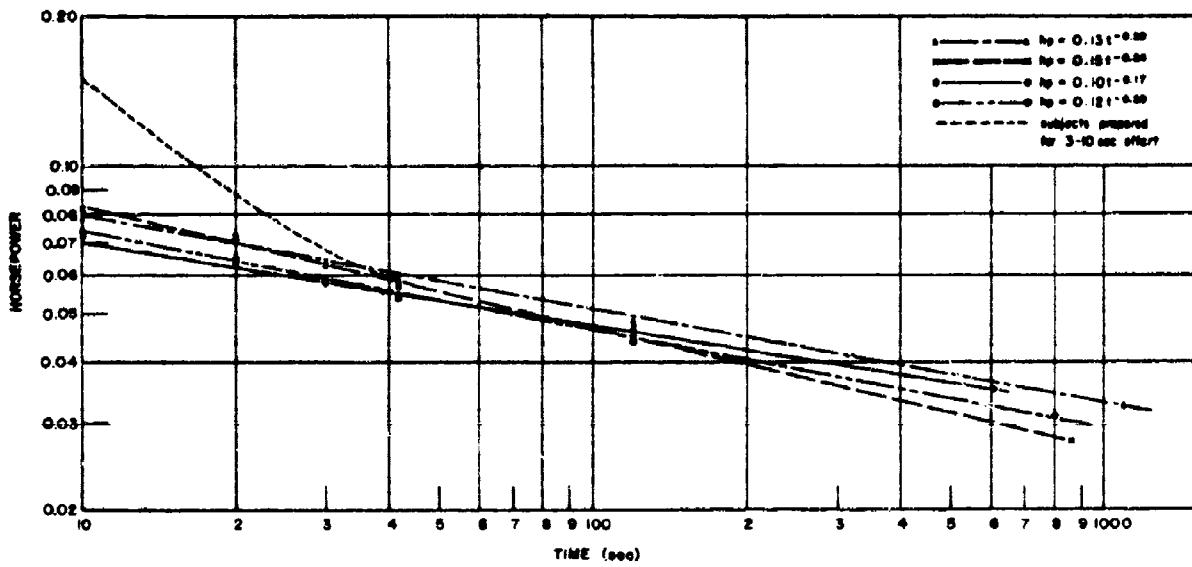
Fig. 1 — Power Generated in Hand Cranking as Influenced by Subject's Expectation of Task Length. (Source: Ref. 7)
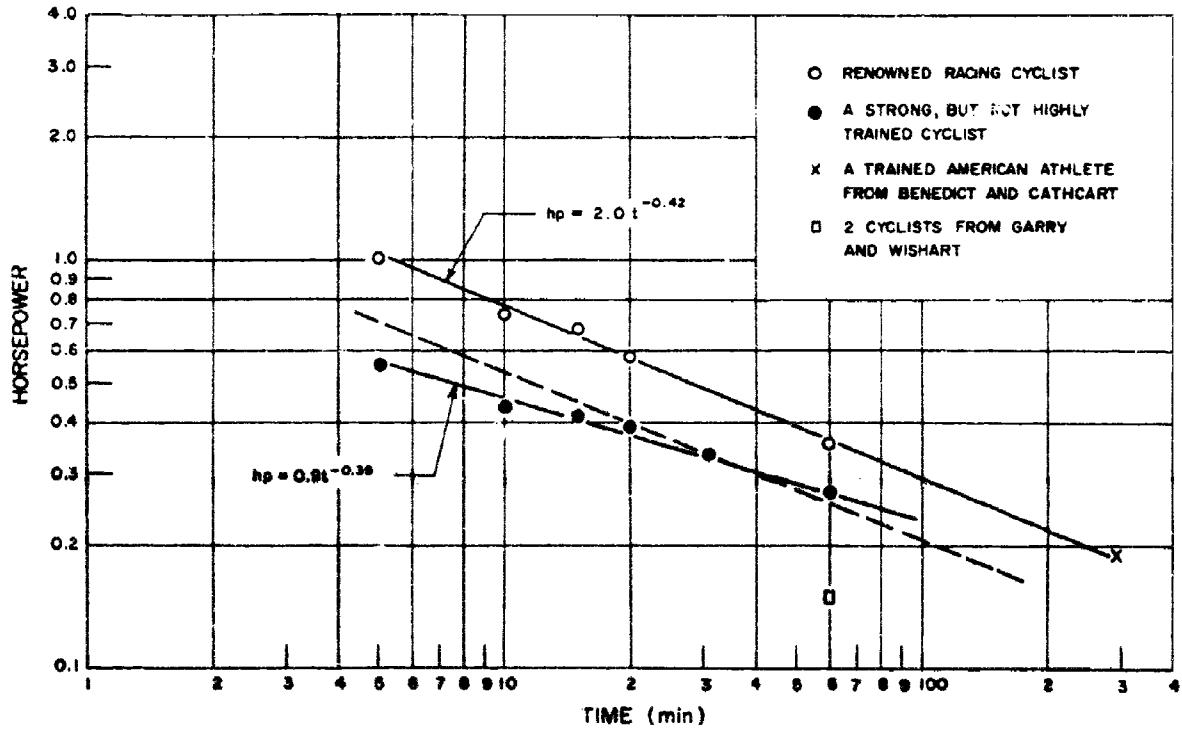

Fig. 2 — Long Duration Pedalling Power. (Source: Ref. 7)

value of some parameter as a function of this representative quantity. As the volume and pressure specifications of the blower are such (as it will be seen later) that 80% blower efficiency would require 0.13 HP from the human operator, blower efficiency was chosen as the parameter that has to be optimized (i.e. to be maximized). The quantity to represent various blower types is the <u>specific speed</u>.

1) <u>Importance of the Concept of Specific Speed.</u>

The concept of specific speed in hydraulic machinery was introduced in 1915 but it became useful for design purposes only very gradually: it took a large amount of published experimental data to obtain the necessary design charts; e.g. Ref. 5 became publicly available in January 1962. The importance of the concept may be characterized by the following statement of a recent authoritative textbook: "The important pump design and performance characteristics are so closely connected with the specific speed that it is impossible to discuss certain features without reference to it." (Ref. 2)

2) <u>Qualitative Considerations Regarding Specific Speed</u>:

The following <u>analogy</u> might be considered: Suppose a duct has to be designed to transport liquid at a given rate along a known distance. In order to pick the most suitable design one has to be able to predict the pumping power required to move the liquid - even before the design has started. As the mathematical relationship between <u>all</u> the possible variables and the power-requirement is not known, answers have to be based on experience with "similar" designs. Similarity of pipelines is based upon equality of a <u>set</u> of dimensionless quantities like the ratio

of length and hydraulic diameter $\frac{L}{D_h}$, Reynolds number, etc. Charts based on the accumulated experience of many decades provide the relationship between the dimensionless quantities. It is possible sometimes to assign physical significance to some of those quantities: e.g. $\frac{L}{D_h}$ is the head loss in ft. of liquid in a pipe with a friction factor of unity carrying fluid at a velocity-head of unity, but nobody thinks of $\frac{L}{D_h}$ this way. It is considered a "type number" describing the duct. The fact, that the set of dimensional quantities characterizing a pipeline is equivalent mathematically to a set of an equal number of dimensionless quantities, does not mean that the set to be used in solving a practical problem has to consist entirely of either dimensional or dimensionless numbers. It is necessary, however, that the dimensional quantities, usually originating from the fixed circumstances of the problem, should be able to define those dimensionless quantities which are not used.

Once the duct is chosen it may be asked: could another "type" of duct be made more efficient? Some reduction in losses for any type duct can be made by improving the smoothness of the walls; however, the general level of performance (head-loss) will be established as soon as the geometry has been chosen on the basis of the type number. Or, if the general level of performance is prescribed then the type-number can be established from the charts compiled from experimental data. Theoretical justification of the above procedure is well known from dimensional analysis. The same type of reasoning applies to the case of fluid-handling equipment. In this case, specific speed will be one of the type-numbers, and other dimensionless variables will enter the calculation.

3) <u>Quantitative Discussion of Specific Speed.</u>

The following reasoning is based on Ref. 3. Characteristic variables of hydraulic machinery are

$$Q, \ n, \ H, \ \text{and} \ D \qquad\qquad 1)$$

i.e. capacity, speed of rotation, head, and a characteristic length indicative of the physical dimensions of the machine. In a formal way one could write

$$H = f_1 \ (Q, \ n, \ D) \qquad\qquad 2)$$

Consider, first, machines that depend upon the inertia of the fluid in order to generate head. This condition automatically involves the fact that the machine will be operated at high Reynolds numbers, where the importance of viscous effects as compared to inertia effects sharply decreases. The above set of four variables does not contain, therefore, the viscosity of the fluid. If the generated head, H, is not high enough to change the density of the medium appreciably all the variables connected with heat and heat transfer can be omitted, provided velocities are far enough from the sound velocity. The above equation involving four variables, therefore, should be adequate for the time being. Consider first one of the possible <u>related sets</u> of Q, H, and n values, namely those of the design point (best efficiency). As H is usually defined in ft of head, same unit as D, it is more convenient to use a corresponding velocity (ft/sec): $\sqrt{2gH}$ instead.

In the above <u>four</u> variables (Q, n, $\sqrt{2gH}$, and D) only <u>two</u> units of measurement are involved: those of length and time. Dimensions of the variables are

$$\text{ft}^3 \ \text{sec}^{-1}, \ \text{sec}^{-1}, \ \text{ft sec}^{-1} \ \text{and} \ \text{ft}.$$

If <u>two</u> units of measurement are involved then <u>three</u> variables can be combined into dimensionless groups while altogether <u>four</u> different groups can be formed, all of them containing three of the four variables.* These four groups will be

(*)Note: In case of four variables and two units of measurement the minimum number of <u>independent</u> dimensionless variables is $4-2=2$.

A1-6

π_1, π_2, π_3, π_4, (π for "Product")

$$\pi_1 = Q \, n^{\alpha} \, (\sqrt{2gH})^{\beta} \qquad \text{3a)}$$

$$\pi_2 = n \quad (\sqrt{2gH})^{\gamma} \, D^{\delta} \qquad \text{3b)}$$

$$\pi_3 = \quad (\sqrt{2gH}) \, D^{\epsilon} Q^{\zeta} \qquad \text{3c)}$$

$$\pi_4 = D \, Q^{\eta} \, n \qquad \text{3d)}$$

In the first equation the dimensions of the variables will be

$$ft^3 sec^{-1} . \quad (ft \, sec^{-1})^{\beta} \, sec^{-\alpha} = ft^{3+\beta} \, sec^{-1-\alpha-\beta} \qquad \text{4)}$$

But the group has to be dimensionless, therefore,

$$3 + \beta = 0 \qquad \text{5a)}$$

$$-1 - \alpha - \beta = 0 \qquad \text{5b)}$$

and so

$$\alpha = +2$$

$$\beta = -3$$

$$\pi_1 = \frac{Q n^2}{(2gH)^{3/2}} \qquad \text{6)}$$

But a dimensionless group raised to any power remains dimensionless: specific speed will be defined as

$$n_s = \pi_1^{\frac{1}{2}} = \frac{1}{(2g)^{3/4}} \frac{n \, Q^{\frac{1}{2}}}{H^{3/4}} \qquad \text{7)}$$

The quantity n_s thus obtained is actually more characteristic of a machine than any of its physical dimensions, as it refers to its performance. Consequently, in the case of an existing machine, its n_s will not change regardless of what speed it is running, provided it operates at the <u>same</u> <u>point</u> of its characteristic curve. ("Same point" could be defined as same percent of maximum flow obtainable at a given speed.)

The other dimensionless quantities of the set can be obtained by similar calculations:

$$d_s = \pi_3^{\frac{1}{2}} = D \frac{H^{\frac{1}{4}}}{Q^{\frac{1}{2}}} (2g)^{\frac{1}{4}} \qquad 8)$$

is called specific diameter; the other two

$$\pi_2 = \frac{nD}{(2gH)^{\frac{1}{2}}} \qquad 9)$$

and

$$\pi_4 = \frac{Dn^3}{Q^{1/3}} \qquad 10)$$

have no specific name. The lack of emphasis on π_2 and π_4 is understandable if one considers that

$$\pi_2 = n_s \, d_s \qquad 11)$$

and

$$\pi_4 = d_s \, n_s^{1/3} \qquad 12)$$

which statements can be readily verified from the formulae derived from d_s and n_s. Furthermore, the fact that Q, H, and n are sufficient to define all the four dimensionless variables n_s, d_s, π_2, and π_4, clearly indicates that absolute physical dimensions of the machine represented by D in the original set of four variables are of secondary importance. (This statement is, of course, subject to the restrictive assumptions made before). This in turn means that, within the limitations of the assumptions, every existing machine can be represented by one point of a $d_s - n_s$ plane. As stated at the beginning of this paragraph, everything here refers to the best efficiency point of the characteristic curve. Therefore, the $d_s - n_s$ charts represent best efficiencies obtained up till now.

It is obvious that certain types of fluid handling machinery are best suited for certain types of requirements. Large diameter, high peripheral speed e.g. is necessary for high heads but would increase flow losses unnecessarily in the case of low head requirement and large flow rates. It would be expected, therefore, that turbomachinery of a particular type (e.g. axial) with the _same_ efficiency would fall in the _same_ region of the $d_s - n_s$ plane. All available evidence supports this assumption and Fig. 3 was prepared on this basis. (Note: N_s of the chart does not use a consistent system of units, omits 2g etc.; therefore, N_s used in this text differs by a constant factor from n_s.)

Traditional definitions of the specific speed (e.g. "revolutions per minute to produce 1 gpm at 1 ft head with a similar machine," etc.) refer to the _physical meaning_ of the specific speed but nevertheless do not bring out its full importance; the specific speed has to be looked upon as a _type-number_ rather than a physical _quantity_.

The $n_s - d_s$ charts that predict maximum obtainable efficiencies for various types of flow-handling equipment on the basis of specific speed obviously make use of experience accumulated in the past and,therefore,do not preclude the possibility of obtaining higher efficiencies as a result of successful development work. In fact, one of the most important features of the $n_s - d_s$ chart is its usefulness in predicting the chances of improvement. E.g. what improvement could be obtained in a so-called "paddle-wheel" blower used by a blacksmith? These blowers are centrifugal-type machines: Fig. 3 uses the expression "Radial." As the blacksmith needs large quantities of air against relatively _small_ resistance a well-developed forge-blower should have a relatively high specific speed (V_1 is large, H_{ad} small: the formula for n_s (Eq. 7) indicates high n_s values). As the shelter-blower has to cope with occasionally _high_ resistances it falls naturally to the lower end of the N_s range of radial blowers. It would, therefore, be necessary to raise the efficiency of the forge-blower _above_ the maximum of its class (0.8) in order to achieve
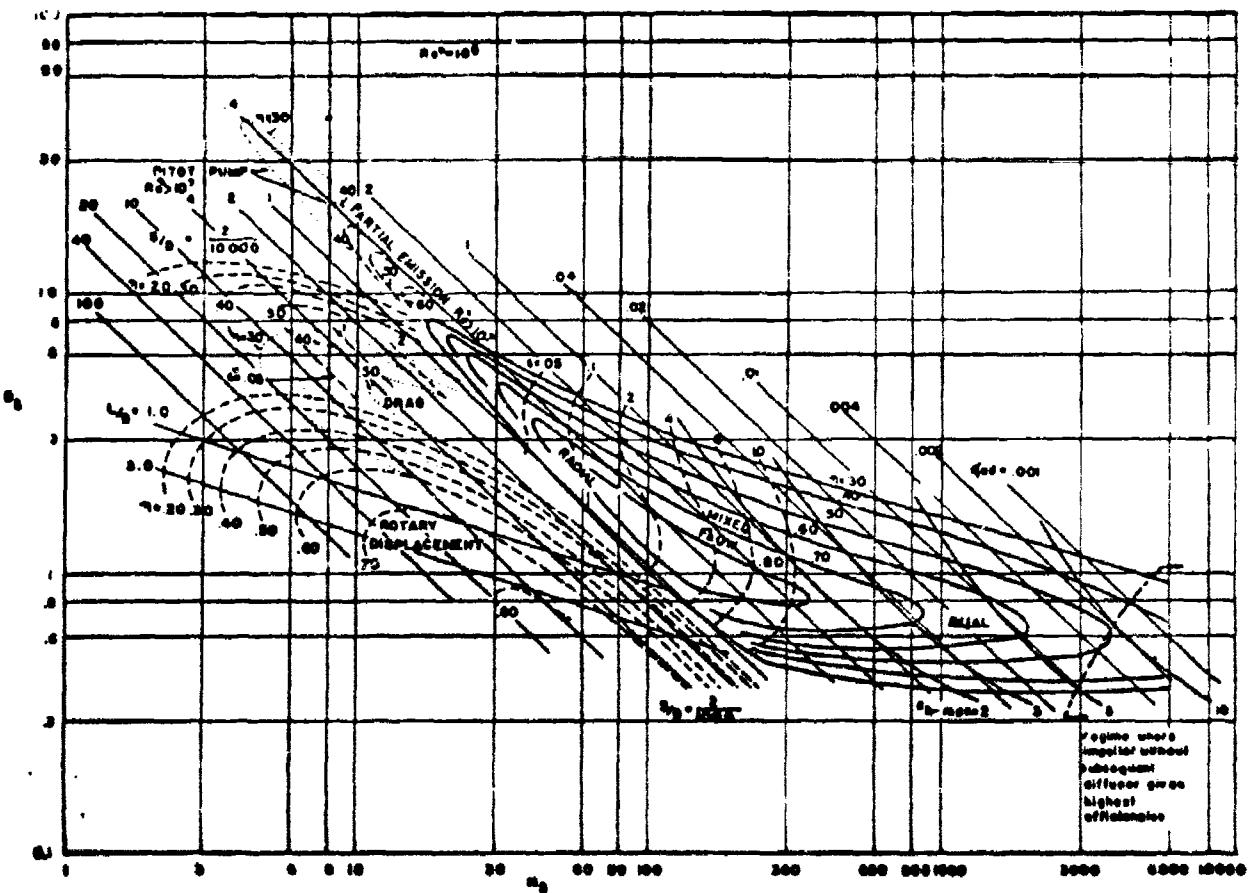
Fig. 3 - Approximate $H_3^{1/3}$ Diagram for Single-Disk Pumps and Low-Pressure
Ratio Compressors. (Source: Ref. (3))

the same 0.8 efficiency at an n_s-d_s point of the chart where the forge-blower does not belong.

It should be mentioned here that until very recently positive displacement machines were not included in the specific-speed charts. It is shown in Baljé's paper (ref. 5) that, though the head in a rotary displacement machine is independent of the rotative speed and the diameter (which is easy to see without resorting to mathematical considerations), the volume flow, however, is affected by the leakage, which depends, among other things, on clearances, the latter depending on the diameter. As the volume flow enters into the expression for efficiency, this latter becomes a function of rotative speed and diameter also. As the leakage flow is a phenomenon observing the laws of fluid mechanics - consequently, flow-similarity - it should be possible to incorporate rotary displacement machines in the same charts as axial and radial fluid machinery. According to Baljé and other writers, the analysis of available data shows that efficiency of positive displacement rotary blowers is a unique function of their geometry expressed in terms of the same n_s and d_s as were explained above. This is the justification of using Baljé's chart, see Fig. 3.

4) **Application of the Specific Speed Concept to the Problem of Selection of the Manual Blower**

Specifications of the Manual Blower are as follows:

1) Manual Operation (emergency conditions): 150 CFM, 4-1/2 in. w. (assuming two 75 CFM filters in parallel): pressure-drop due to the ductwork: 2-1/2 in. w., (parabolic function of the flow rate); pressure drop across the filter: max. 2 in. w., (linear function of the flow rate).

2) Motor operation: 300 CFM. Pressure drop was not specified for this mode of operation but it is assumed that the filters will be bypassed. This latter condition sets the pressure drop of the system at $\left(\frac{300}{200}\right)^2 \times 2.5 = 5.6$ in. w.

It follows from 1) that the theoretical power requirements (100% efficiency) are 0.1 HP (approx.) Specific speed (N_s) of the blower will be computed on the basis of the new specifications as follows:

$$N_s = N \frac{V_1^{1/2}}{H_{ad}^{3/4}} = N \frac{\left(\frac{150}{60}\right)^{1/2}}{\left(4.5 \frac{5.18}{0.07}\right)^{3/4}} = \frac{N}{48.6}$$

and

$$H_{ad} = \frac{\Delta p}{\gamma}$$

where

$V_1 = 150$ CFM

$\Delta p = 4.5$ in. w.

$\gamma = 0.07$ lb/ft^3

N: speed in RPM

H_{ad}: adiabatic head, ft-lb/lb

It was pointed out that the best speed obtainable by a human operator is 50 RPM, approx. Furthermore, the specific speed range for best efficiency (80%) for various types of blowers (Fig. 3) designates the N_s arrived at above which in turn specifies the speed of the blower based on H, Q requirements: $N = 48.6\ N_s$. The actual RPM's will be:

14600 for the axial machine (minimum $N_S = 300$)
2820 for centrifugals (minimum $N_S = 60$)
1460 for rotary positive blowers (minimum $N_S = 30$)

In case of motor drive (300 CFM) the speed has to be twice the above values.

Considering the case of the human operator (N = 20 RPM) the following speed-changes have to be obtained: 1:29.2 for the rotary positive machine, 1:56.6 for the centrifugal blower and 1:292 for an axial machine. This latter figure rules out the axial machine at once: a speed change of 292 would be very inconvenient. As for the centrifugal machine, a 1:56.6 speed-change could be achieved in two steps (e.g. 7.55 x 7.55), but gears of 1:7.55 ratio require careful machining in order to obtain high efficiencies and have to be carefully aligned. In short, the machine would be more expensive than a blower where only a conventional bicycle-chain drive is needed, providing in a convenient single step the required 1:4 speed change. This ratio is well within the limits of the chain drive.

In addition to the two-step gear drive there is another consideration in connection with the centrifugal blower. The specific speed at the point of best efficiency designates the geometry of the machine, and this geometry will determine the pressure versus volume curve of the blower. (See e.g. Ref. 2, p.162, Fig. 9.1)
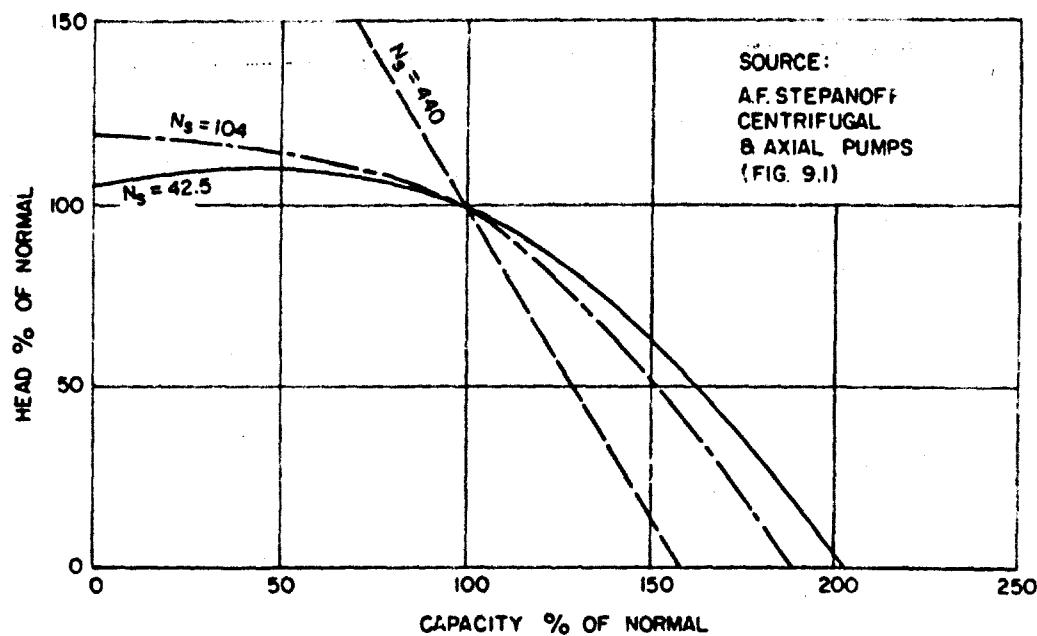

Fig. 4 - Head-Capacity Curves for Several Specific Speeds

Fig. 4 was prepared on the basis of Stepanoff's survey. It can be seen that centrifugal blowers with $N_s = 42.5$ have an inherently unstable part of their characteristic curve. If two or more blowers have to be used in a shelter working in parallel on the same duct system, the well known difficulties of "pumping" may arise quite easily. In order to get constantly rising characteristic curves, at least $N_s = 75$ has to be applied. According to the equation quoted above, this means N = 3650 RPM in the case of the human operator or 7300 RPM for motor-drive. Accordingly, a speed-change of 73 would be necessary rather than 56.5 as calculated before.

It should be mentioned here that the expression "centrifugal" blower in the above considerations included all types of centrifugal machines regardless of how their blades are constructed (including, therefore, the so-called radial blowers with straight blades, provided

the latter operate on hydrodynamical principles.) Having thus considered the "hydrodynamic" type machines the positive displacement blowers will follow.

Consider first the rotary-positive blowers. If top efficiency has to be obtained with the Roots-blower careful manufacturing procedures are needed: e.g. clearances of $\frac{1"}{100}$ have to be maintained between runners and housings. (This clearance requirement too can be obtained from Fig. 3: $S/D = \frac{2}{1000}$ and $D_s = 0.8$ result in the above $\frac{1"}{100}$ value.) In order to have good mechanical efficiency the timing gears of the blower have to be carefully machined. It is known, finally, that the Roots blower needs careful alignment if good performance has to be maintained; a remote blast may shake the blower enough to affect its performance.

It seems, therefore, that a reciprocating, positive displacement, slow machine would be the best solution.

D. CONCLUSION

The procedure used for selection seems to favor the positive displacement type machine; it has to be considered, however, that the centrifugal blower is penalized by the need of the speed-changer.

It should also be mentioned that the positive-displacements pump has the advantage of a constant-volume characteristic; this feature could be important in an emergency.

It is felt, therefore, that the manual blower for the application discussed in this Report should be a positive displacement machine.

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APPENDIX 2

OUTPUT OF POSITIVE DISPLACEMENT AND CONSTANT DELIVERY BLOWERS

The useful output of a constant delivery rate blower (W_{cd}), such as a centrifugal blower, can be calculated from the pressure difference created by the machine (Δp_{cd}) and the delivery rate (V):

$$W_{cd} = \Delta p_{cd} V$$

In the case of a positive displacement blower, however, the time average (W_{av}) of the constantly changing, instantaneous power output (W) will be more than $\Delta p_{cd} V_{av}$; the excess power depends on the resistance law of the system served by the blower. Assume a quadratic resistance law (orifice type).

A. Excess Power Caused by Variable Delivery Rate (no losses in the blower)

The Scotch-yoke type drive of the blower and the small compressibilty effects permit the following expression for the instantaneous delivery rate V:

$$V = V_o \sin \omega t \tag{1}$$

where

$$0 \leq t \leq \frac{\pi}{\omega}$$
$$V_{av} = \frac{\omega}{\pi} \int_0^{\pi/\omega} V_o \sin(\omega t) \, dt = \frac{2}{\pi} V_o \tag{2}$$

Assuming a quadratic resistance, the instantaneous pressure is

$$\Delta p = cV^2 = cV_o^2 \sin^2(\omega t) \tag{3}$$

and

$$\Delta p_{av} = cV_o^2 \frac{\omega}{\pi} \int_0^{\pi/\omega} \sin^2(\omega t) dt = c \frac{V_o^2}{2} = \frac{\pi^2}{8} cV_{av}^2 \tag{4}$$

From equations 2, 3, and 4

$$C = 2\frac{\Delta p_{av}}{V_o^2} = \frac{8}{\pi^2} \frac{\Delta p_{av}}{V_{av}^2} \tag{5}$$

Of the above derived quantities, V_{av} is the same for both the constant delivery rate machine (e.g. a centrifugal blower) and the variable delivery rate machine (such as the positive displacement blower). One can see from Equation (4)

$$\Delta P_{cd} = \frac{8}{\pi^2} \Delta P_{av} \qquad (4a)$$

i.e the same flow rate (V_{av}) and throttle geometry (c) will result in a lower average pressure $(\Delta p)_{cd}$ for the constant delivery machine. It should be noted here that a damped manometer, such as the one used in the test, will indicate Δp_{av} for a positive displacement machine.

The theoretical power requirement (W_{th}) for a positive displacement machine will be calculated now.

From Equations 1, 2, 4, and 5 the instantaneous power requirements will be:

$$W_{th} = V_{\Delta p} = c V_o^2 \sin^2 \omega t \ V_o \sin \omega t = \Delta P_{av} V_{av} \sin^3 \omega t \qquad (6)$$

The average power will be

$$W_{av} = \frac{\omega}{\pi} \int_0^{\pi/\omega} W_{th} dt = \Delta P_{av} V_{av} \ [\frac{\cos^3 \theta}{3}]_{\theta=0}^{\theta=\pi} = \frac{4}{3} \Delta P_{av} V_{av} \qquad (7)$$

In a constant delivery rate blower, from Equation 4a

$$W_{cd} = (\Delta p)_{cd} V_{av} = \frac{8}{\pi^2} \Delta P_{av} V_{av} \qquad (8)$$

therefore

$$\frac{W_{av}}{W_{cd}} = \frac{4 \pi^2}{3.8} = \frac{\pi^2}{6} \text{ (for the same } V_{av}) \qquad (9)$$

The maximum possible gain using an ideal surge tank will be

$$\frac{\Delta P_s}{W_{cd}} = \frac{W_{av} - W_{cd}}{W_{cd}} = \frac{\pi^2}{6} - 1 = 1.64 - 1 = 0.64 \qquad (10)$$

If an ideal surge tank could be built, the theoretical power requirements of a positive displacement machine equipped with the ideal surge tank would be the same as that of a constant delivery rate blower.

Because of the fact that the surge tank has to change its volume, some energy will have to be spent on operating the surge tanks. If the surge tanks are correctly designed, this energy will not be a significant percentage of W_{av}. In the case of such a "quasi constant delivery" blower $W_{qcd} \approx W_{av}$.

The theoretical energy requirements explained above for different blowers make it necessary to define the blower efficiency more exactly.

Assuming a system with a square-law resistance (as we did before) the efficiency of a constant delivery-rate blower (e.g. a centrifugal fan) will be

$$\eta_{cd} = \frac{\Delta p V}{W_{in}} \tag{11}$$

When the same quadratic-law system is coupled with a positive displacement machine, the damped manometer will show a higher average pressure (ΔP_{av}) at the same delivery rate: $V_{av} = V$. Δp_{av} can be calculated from Equation 4a, or, if Δp_{av} is known from tests (as was the case in the investigations reported here) then Δp_{cd} can be calculated from Equation 4a. Therefore, if direct comparison has to be made between a positive displacement and a constant delivery rate machine, both applied to the same square-law resistance system, the efficiency of the positive displacement machine will be

$$\eta = \frac{8}{\pi^2} \frac{\Delta p_{av} V_{av}}{W_{in}} \tag{12}$$

This η can be compared directly with the efficiency of the constant delivery machine (Equation 10). If the factor $\bar{\delta}/\eta^2$ is not used, the resulting quantity

$$\eta^* = \frac{\Delta P_{av} \; V_{av}}{W_{in}} \qquad (13)$$

should not be considered as the "efficiency" of the positive displacement machine; in this report, we call it the "effectiveness" of the blower. In Table II, this effectivness is shown. This "effectiveness" is less than the mechanical efficiency (η_m) of the positive displacement blower: from Equation 7

$$\eta_m = \frac{4}{3} \frac{P_{av} \; V_{av}}{W_{in}} \qquad (14)$$

Therefore, from Equations 11, 12, 13 and 14

$$\eta = \eta_{cd} < \eta^*_m < \eta_m \qquad (15)$$

In evaluating test results of a positive displacement blower, the quantity η_m is very useful: it derives from directly measured quantities (ΔP_{av} is shown on the damped manometer) and shows how good the components of the machine are: in the case of valves without pressure losses, in the absence of friction and leakage, η_m would be 100% in a quadratic resistance system. It can be seen from Equation 13, that "effectiveness" (η^*) differs only by the constant 4/3 from η_m. In Figures 15 and 16, η^* was plotted, as the measured quantities P_{op} and P_{in} are immediately available on the graph. P_{op} is also useful in calculating immediately ΔP_s (Equation 7):

$$\Delta P_s = \frac{P_{op}}{3} \qquad (16)$$

Note: When power is in other units than HP, the symbol W is used. W' or P stands for power in HP.

B. How Much is the Loss in Efficiency if Δp_s Fluctuation is Tolerated?

Some energy has to be sacrificed in order to make the surge tank change its volume. This energy comes most conveniently from the pressure: a small pressure fluctuation Δp_s will be allowed. (Of course, $\Delta p_s < \Delta p_{av}$.)* The Δp_s pressure fluctuation will cause also a small fluctuation in the delivery rate: V_s.

We can write

$$\Delta p = \Delta p_{av} + \Delta p_s \sin(\omega t) \tag{17}$$

and

$$V = V_{av} + V_s \sin(\omega t) \tag{18}$$

From Equation 3 and Equation 17, the instantaneous pressure can be written

$$\Delta p = cV^2 = c\,[V^2_{av} + 2V_{av}\,V_s \sin(\omega t) + (V_s \sin(\omega t))^2] \tag{19}$$

where $(V_s \sin(\omega t))^2$ will be omitted as a small quantity of secondary order. From Equations 17 and 19

$$\Delta p_s \sin(\omega t) = \Delta p - \Delta p_{av} = c\,2V_{av}\,V_s \sin(\omega t) \tag{20}$$

Instantaneous power will be (from Equations 17, 18, and 10)

$$W = V\Delta p = (V_{av} + V_s \sin(\omega t))(\Delta p_{av} + \Delta p_s \sin(\omega t)) = \Delta p_{av}\,V_{av} +$$

$$(V_{av}\,\Delta p_s + V_s\,\Delta p_{av}) \sin(\omega t) + \Delta p_s\,V_s \sin^2(\omega t) \tag{21}$$

Taking the time average of the instantaneous power, only two members of Equation 21 remain, as $\sin(\omega t)$ has a time average of zero.

$$W_{av} = \int_0^t W\,dt = \Delta p_{av}\,V_{av} + \frac{\omega}{2\pi}\,\Delta p_s\,V_s \int_0^{\frac{2\pi}{\omega}} \sin^2(\omega t)\,dt$$

$$= \Delta p_{av}\,V_{av} + \frac{\Delta p_s\,V_s}{2} \tag{22}$$

*It should be noted that Δp_{av} will be less than in the case of no surge tank, but somewhat larger than Δp_{cd}.

Rearranging Equation 21 we obtain the power lost because of

fluctuation, Δp_s:

$$W_{av} - p_{av} V_{av} = \frac{\Delta p_s V_s}{2} \tag{23}$$

but V_s has yet to be calculated.

Let

$$\Delta p_{av} + \Delta p_s = c(V_{av} + V_s)^2 \approx c V^2_{av} + 2cV_{av}V_s \tag{24}$$

from which

$$2cV_{av}V_s = \Delta p_{av} + \Delta p_s - cV^2_{av} \tag{25}$$

and

$$V_s = \frac{\Delta p_{av} + \Delta p_s - c V^2_{av}}{2cV_{av}} \tag{26}$$

If Δp_s is small (i.e. the surge tank operates fairly well),

$$\Delta p_{av} \cong cV^2_{av}$$

and

$$V_s = \frac{\Delta p_s}{2cV_{av}} \tag{27}$$

Combine Equations 27 and 23

$$W_{av} - \Delta p_{av} V_{av} = \frac{\Delta p_s}{2} \frac{\Delta p_s}{2cV_{av}} \frac{\Delta p_s^2}{4cV_{av}} \tag{28}$$

The decrease in efficiency due to Δp_s will be

$$\Delta \eta_s = \frac{W_{av} - \Delta p_a V_{av}}{p_{av} V_{av}} \frac{\Delta p_s^2}{4cV_{av} \Delta P_{av} V_{av}} \tag{29}$$

But

$$c = \frac{\Delta p_{av}}{V^2_{av}} \tag{30}$$

and therefore

$$\Delta \eta_s = \left(\frac{\Delta p_s}{2\Delta p_{av}} \right)^2 \tag{31}$$

It can be seen from Equation 31 that $\Delta p_s = 1$ in. w. will result in

$\Delta \eta_s = 1.23\%$ for $\Delta p_{av} = 4.5$ in. w.

Consequently, it should be possible to build a surge tank that would cause the positive displacement blower to operate with the quadratic resistance system practically at the same efficiency as a constant delivery machine.

If the system connected with the blower obeys a power law with an exponent less than 2, η_m and η^* will be closer to η than in a square-law system. Filters used in connection with the present application are elements that have a less-than-quadratic resistance curve. This possibility was not investigated during the program in a quantitative manner.

3.0 Volume of Surge Tank

When the instantaneous flow rate (V) is larger than V_{av}, it will be stored in the surge tank. The surge tank will release the stored air when $V > V_{av}$.

The instantaneous quantity to be stored, s, can be written from Equations 1 and 2:

$$V - V_{av} = \frac{\pi}{2} V_{av} \sin(\omega t) - V_{av} = (\frac{\pi}{2} \sin(\omega t) - 1) V_{av} \quad (32)$$

The quantity to be stored from t = 0 to t = t_s will be

$$S = \int_0^{t_s} s\, dt = \int_0^{t_s} (V - V_{av})\, dt = V_{av} \int_0^{t_s} (\frac{\pi}{2} \sin(\omega t) - 1)\, dt$$

$$= \frac{V_{av}}{\omega} \frac{\pi}{2} (1 - \cos(\omega t_s)) - V_{av} t_s \quad (33)$$

At t - t_s, when s = 0, no storage is needed any more.

t_s can be calculated from Equation 32:

$$\frac{\pi}{2} \sin (\omega t_s) - 1 = 0$$

$$\omega t_s = 0.7 \text{ rad} = 40° \tag{34}$$

Therefore

$$S = \frac{V_{av}}{\omega} \left[\frac{\pi}{2} - \frac{\pi}{2} \cos 0.7 - 0.7 \right] = 0.34 \frac{V_{av}}{\omega} \tag{35}$$

If Q is the volume of the cylinder, the displaced volume can also be written (for a double-acting machine)

$$V_{av} = 2 \frac{\omega}{2\pi} Q \tag{36}$$

and from Equation 13

$$S = \frac{0.34}{\pi} Q = 0.108 Q \tag{37}$$

In our case, Q = 0.5 ft^3, therefore S = 0.054 ft^3 = 93 in^3

APPENDIX 3

ANALYSIS OF TEST RESULTS

The results shown in the body of the Report in Tables I and II were plotted in Figure 15 for the blower with the surge tank and in Figure 16 for the blower operating without the surge tank. In the analysis of the results, the curves of Figures 15 and 16 will be used rather than the actual test points.

The ultimate objective of this analysis is to find out whether the blower really approximates a constant delivery machine to such an extent that in Table I, it is justified to use "efficiency" instead of "effectiveness". As discussed in Appendix 2, "efficiency" can be used in the same manner as for a constant delivery machine if ΔP_s is significantly reduced by the surge tanks and $\Delta \eta_s$ is small (Equation 31).

In order to find ΔP_s, the other losses have to be found first. The calculation will be performed for the design pressure: $\Delta p = 4.5$ in. w, 110 rpm (Fig. 15). Assume, that the blower is close to being a constant delivery rate machine (this has to be proved from the calculations) $\Delta p \cong \Delta p_{av}$. In this case, the equivalent "positive displacement" performance (i.e. surge tanks immobilized) will be (Equation 4a)

$$\Delta p_{av} = \frac{\pi^2}{8} \Delta p_{cd} = 1.23 \cdot 4.5 = 5.5 \text{ in. w.}$$

The total amount of losses for $\Delta p_{av} = 5.5$ in. w. will be $\Delta P = 0.0365$ HP, from Fig. 17. (Figure 17 was prepared from Figures 15 and 16, using the faired-in input and output curves).

From Figure 16, P_{op} for 5.5 in.w. will be 0.07 HP; therefore, from Equation 16, $\Delta P_s = \frac{0.07}{3} = 0.023$ HP. ΔP can be written as $\Delta P = \Delta P_s +$ $+ \Delta P_L + \Delta P_f + \Delta P_V$ where the individual losses are the sinusoidal

excess power, leakage loss, friction and valve losses, respectively. As

ΔP_s is known, the losses due to leakage will be calculated now for

$V = 82.5$ cfm, and $\Delta p = 5.5$ in. w. The volumetric inefficiency, $V = V_{th} - V$,

is caused by air from leakage, but also by an air volume that is compressed

into the "clearance volume" of the blower (the volume in the box, additional

to the working cylinder). This volume of air in the clearance is compressed

at the beginning of every stroke and released at the end of the stroke, there-

fore it does not contribute to the power losses (it acts like a spring). As

the clearance volume is approximately 14 ft^3, and the working volume of the

blower 1 ft^3, it can be calculated that for a pressure differential of 4.6 in. w,

only 90% of the theoretical air-delivery can be expected; for 6.0 in. w, only

85%. As the displacement for every turn of the flywheel was 1 ft^3, the volumetric

deficiency due to the clearance volume can be written as $V_c = n - 0.87n$ for

n rpm and $\Delta p = 5.5$ in. w. Consequently, the volumetric losses due to leakage,

ΔV_L, will be:

$$\Delta V_L = V_{th} - V - V_c = (n - v) - (n - 0.87\, n) = (110 - 83.5)$$

$$- (110 - 96) = 12.5 \text{ ft}^3/\text{min}$$

(Note: $V_{th} = n$ numerically because displacement of the blower is 1.0 ft^3/turn).

This 12.5 ft^3/min. is lost because of leakage, created by a pressure

differential of $\Delta p_t = \Delta p + \Delta p_v$ where Δp is the vacuum created by the blower in

the suction pipe and Δp_v is lost when the flow passes through the valves. At

the present time we do not know how much Δp_v is, therefore, as a first approxi-

mation, take $\Delta p_v = 0$. For $\Delta p = 5.5$ in. w., the power loss due to leakage will be

$$\Delta P_L = \frac{12.5 \times 5.5}{6410} = 0.010 \text{ HP}$$

The next source of losses is the friction between cylinder wall and piston, ΔP_f. (Other mechanical losses were found to be below the accuracy of the measurements, due to low speed and the use of ball bearings). Average piston velocity at 110 rpm and 7-in. stroke is 2.14 fps. Considering the 1.6 lb. (approx.) force needed to move the piston (Section 6.1) the power loss due to friction, ΔP_f will be 0.006 HP.

The losses due to pressure drop in the valves ΔP_v and sinusoidal operation of the blower ΔP_S will be

$$\Delta P_v + \Delta P_S = \Delta P - \Delta P_f - \Delta P_L = 0.021 \text{ HP}$$

This value of 0.021 HP is even less than the calculated value of ΔP_S alone: 0.023 HP, which shows that the assumption of $\Delta P_v = 0$ was justified. The fact that the valve losses for 110 rpm are negligible is also indicated on Figures 15 and 16 where the extrapolated delivery rate for $\Delta p = 0$ is practically the theoretical 110 ft^3/min: if the inlet valves had an appreciable pressure drop, some leakage would occur when $\Delta p = 0$ in the inlet duct. The discrepancy between $\Delta P_S = 0.021$ and 0.023 is too small for the accuracy of the measurements to be traced down.

We assume, therefore, that $\Delta P_S = \dfrac{0.021 + 0.023}{2} = 0.022$ HP; the performance of the blower <u>with the surge tanks</u> will be judged from the decrease in ΔP_S. We assume now, that ΔP_f will be the same for the operation with surge tanks at $\Delta p = 4.5$ in w.: 0.006 HP. ΔP_L has to be calculated because ΔV_L is different at $\Delta p = 4.5$ in.w.

$V_C = n - 0.9n$ for $\Delta p = 4.5$ in. w., therefore $V_C = 11.0$ ft^3/min.

$\Delta V_L = n - V - V_C = 110 - 86 - 11 = 13$ ft^3/min (V was taken from Fig. 15.)

$\Delta P_L = \dfrac{13 \times 4.5}{6410} = 0.009$ HP

From Figure 17, the total amount of losses will be $\Delta P = 0.024$ HP.

As $\Delta P_V = 0$, $\Delta P_S = 0.024 - 0.009 - 0.006 = 0.009$ HP

From Equation 29 and taking $P_{op} = 0.06$ HP from Figure 15,

$\Delta \eta_s = \dfrac{0.009}{0.06} = 0.15.$

From the ΔP_S value for the operation without the surge tank we

obtained previously that $\Delta P_S = 0.022$ HP; the decrease in the power loss due

to the sinusoidal operation is therefore $\dfrac{0.022 - 0.09}{0.022} = 59\%.$

As calculated in Appendix 2, a fluctuating pressure difference of

$\Delta p_S = 1.0$ in. w. would cause only $\Delta \eta_s = 1.2\%$ instead of 15%. As $\Delta \eta_s$ is

not negligible, in this case we are confronted with effectiveness and not

efficiency. A similar calculation for $\Delta p = 4.5$ in. w. but 145 rpm, yields

very interesting results. ΔP_V is not zero anymore, but 0.0255 HP, equivalent

to a valve pressure drop of 1.4 in. w. This 1.4 in. w. pressure drop is the

sum of the increased vacuum in the casing during inflow because of the pressure

drop across the inlet valves, and the overpressure during the exhaust when the

exhaust valves will cause a pressure rise. Assuming that the two pressure

differences are equal, a vacuum of 0.7 in. w. will prevail in the casing when

the useful output of the machine is $\Delta p = 0$ (measured in the intake duct).

The amount of leakage caused by the 0.7 in. vacuum can be readily seen from

the 110 rpm runs, where there is no difference between duct pressure and

casing pressure during the inflow: $\Delta V = 4$ cfm for the run without the surge

tank and 5 cfm for the run with surge tank. It can be seen from the 145 rpm

runs that the 4-5 cfm leakage agrees quite well with the missing cfm-s at $\Delta p = 0$ for the 145 rpm runs.

Having established ΔP_v for the case without the surge tank, the calculation was continued for the operation with surge tanks, similar to the 110 rpm run explained in detail before. It was found that while the valve pressure drop decreases the efficiency in the 145 rpm runs, the surge tanks operate quite well, increasing the efficiency: ΔP_S is practically zero for the 145 rpm run at $\Delta p = 4.5$ in. w. As explained in Appendix 2, the "effectiveness" of the blower at this point is, therefore, the conventional blower efficiency. Hence the heading "Efficiency or Efectiveness" in Table I.

DOCUMENT CONTROL DATA - R&D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1 ORIGINATING ACTIVITY (Corporate author)	2a REPORT SECURITY CLASSIFICATION
The Franklin Institute, Laboratories for Research & Development, 20th & Benjamin Franklin Parkway, Philadelphia, Pennsylvania 19103	Unclassified
	2b GROUP

3 REPORT TITLE

Manual Blower Development (U)

4 DESCRIPTIVE NOTES *(Type of report and inclusive dates)*

Final Report

5 AUTHOR(S) *(Last name, first name, initial)*

Vermes, Geza nmn; Wachtell, George Peter

6 REPORT DATE	7a TOTAL NO OF PAGES	7b NO. OF REFS
September 30, 1965	51	

8a CONTRACT OR GRANT NO.	9a ORIGINATOR'S REPORT NUMBER(S)
Contract No. OCD-OS-62-280	
b. PROJECT NO. 1400	F-B1983
c. Task No. 1430	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)
d. Work Unit 1432A	

10 AVAILABILITY/LIMITATION NOTICES

Distribution of this report is unlimited

11 SUPPLEMENTARY NOTES	12 SPONSORING MILITARY ACTIVITY
	Office of Civil Defense (OCD) Department of the Army - OSA Washington, D. C. 20310

13. ABSTRACT

This report presents the method of selecting a suitable blower type for manual drive; describes the experimental and theoretical investigations which led to the development of three key components (valves, piston and surge-tank). Results of tests are analyzed that show the blower having more than 65% total efficiency and a stiff pressure vs volume curve. Design for mass production is suggested; calculation of components for the mass produced version is explained so that the mass produced blower should at least be as efficient as the test-version described in this report.

DD FORM 1473 1 JAN 64

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